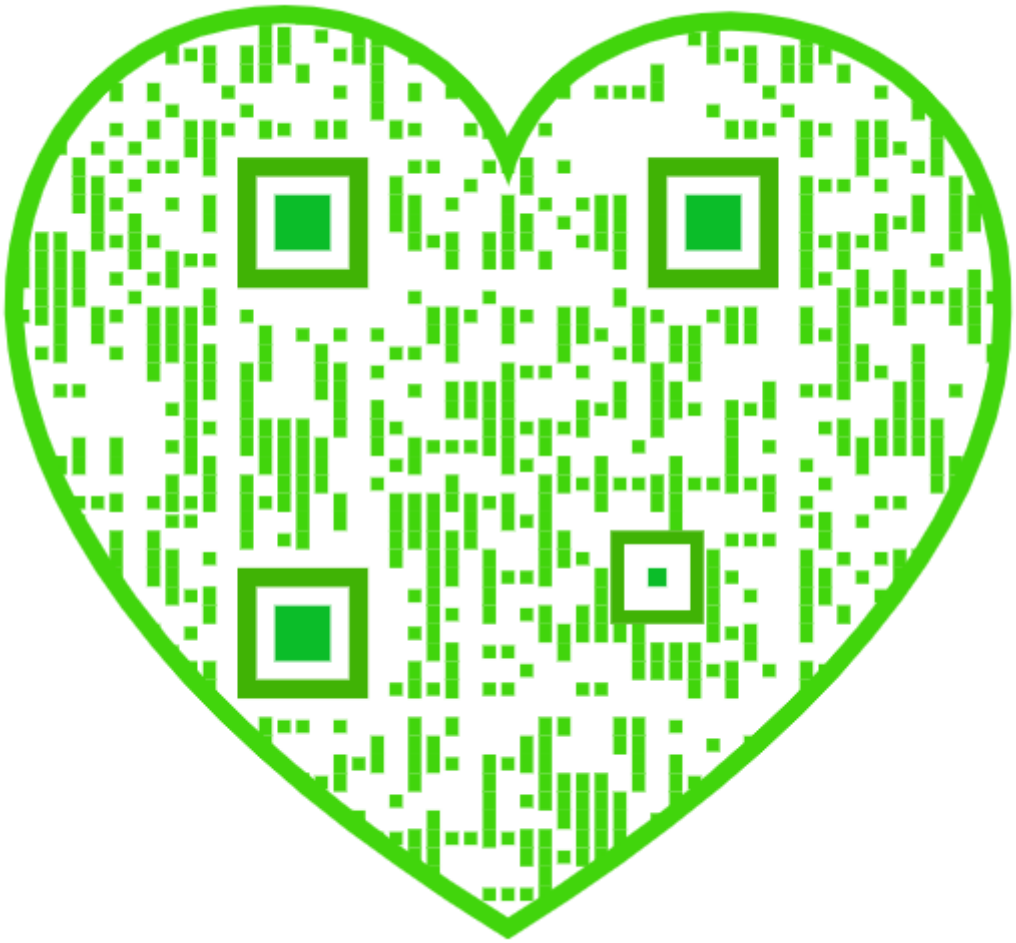

MODERN LOCOMOTIVE VALVES AND VALVE GEARS

By

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By Chas. L. McShane.

DEDICATION.

THIS BOOK IS AFFECTIONATELY DEDICATED TO

THE MEMORY OF MY FATHER,

CHARLES MCSHANE,

WHOSE WRITINGS ON MECHANICAL SUBJECTS

ASSISTED SO MANY MEN, WITH THE HOPE

THAT THIS WORK MAY HELP OTHERS

HAVING TO DO WITH MODERN

LOCOMOTIVE VALVES AND

VALVE GEARS.

THE AUTHOR.

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PREFACE.

While it is true that there have been many good books published regarding the subject of valves and valve gears, it is, however, a regrettable fact that there is not a book on the market today treating the modern types from a strictly practical standpoint.

In attempting to supply this deficiency, we have assumed no knowledge on the part of the reader regarding the subject, and, as a result, have devoted considerable space to elementary, or fundamental principles, so as to enable the apprentice or student to travel the road from early to modern practice with ease. Naturally, some of this information cannot be expected to be of much interest to the experienced man, but the author believes that a statement of the general principles, so compact in form and simple in arrangement that all herein contained can be thoroughly mastered, and become a part of the mental stock of the reader, will be interesting and profitable. In fact, the work is presented with no journalistic pretensions, but special care has been take to use plain, common sense language, so that it may be understood by anyone who can read the English language. But, in the final analysis, the work must speak for itself—its value must be measured by

the merit its use may develop, its faults could not be diminished nor excused by anything the author might here say.

While special efforts have been made to use the utmost accuracy, it is almost impossible to prepare a work of this nature entirely free from error, but the book is presented with a realization of its imperfections. If the reader will kindly call the publishers' attention to any errors, of omission or commission, he may find in the work, correction will be made in future editions, and the favor will be greatly appreciated.

To enumerate all the sources of information to which the author is indebted, in connection with the preparation of this work, would be an almost impossible task, yet we cannot refrain from publicly acknowledging the numerous courtesies, and the aid, extended by the American Locomotive Company and the Baldwin Locomotive Works; and to Mr. John J. Lahey, of the C. C. C. & St. L. Ry. Co., Mattoon, Ill., and Mr. Thomas J. Pembroke, of the Pennsylvania Co., Chicago, Ill., for carefully reading the proofs and for their many valuable suggestions. To other friends and inventors, we have privately made due acknowledgment.

C. L. McS.

Chicago, Ill.,
November 1, 1917.

INTRODUCTION.

In the preparation of this work we have attempted to confine our discussion to the valve and the valve gear used on the locomotive, as distinguished from the stationary engine. Of course, some principles and practices are common to other types, and may be helpful in some instances, but where our statements are in conflict with the approved practice relating to such engines, the fact should be kept in mind that the following pages are primarily intended to apply to locomotive construction and practice only.

It is our desire to make the discussion of the valve, and the valve gear, as plain as the technical character of the subject will permit, but the subjects are most important, and a study of them should not be entered upon, by the apprentice or student, until he has become thoroughly familiar with some of the terms used.

With this idea in mind we shall first explain the meaning of the terms and expressions most commonly used in discussing the valve, and the valve gear, and, later on, give the reason for their application.

This arrangement is somewhat irregular, but it will serve the two-fold purpose of supplying the reader with indispensable knowledge, and also dispense with the repetition of such definitions throughout the work.

General Definitions and Technical Terms.

ADMISSION.—A word used to describe the entrance of steam, from the boiler, through the valve chamber to the cylinder, to be used in moving the piston.

Inside-) Is the term applied when steam from the boiler is supplied to the center of a *piston* valve between its two heads, and is exhausted through the channels at the end of the cylinder. The point is shown in Fig. 54.

Outside-) Is just the opposite of *inside*-admission; that is, it commences when the outside edge of a *slide* valve reaches the outer edge of the steam port, and the valve begins to open the port. This point is shown in Fig. 6.

Point of-) Is the position of the valve when it begins to uncover the steam port for the admission of steam to the cylinder.

-Port.) Is a channel, or port, through which steam from the boiler gains access to the cylinder. See Fig. 2.

Pre-) Is the distance from the front of the advancing piston to the end of the piston stroke when the valve

begins to open for the admission of steam; the space the crank must travel to reach the dead center. It begins when the valve opens the port for lead and ends when the crank is on center—the point where lead commences. See Fig. 10.

-*Valve.*) The valve which controls the admission of steam to the cylinder.

ADVANCE.—(See Angle of Advance, Linear Advance, etc.)

ANGLE OF ADVANCE.—Represents the distance the center of the eccentric is advanced toward the crank-pin from a line drawn through the center of the axle at right angles to the center line of motion. Or, the combined angle of lap and lead through which the valve must move from its central position to the position it should occupy when the crank is on dead center. It is sometimes designated *angular advance* and is indicated by Fig. 17.

Lead-) (See Lead.)

BACK-PRESSURE.—(See Pressure.)

BALANCED-VALVE.—A *slide*-valve is said to be balanced when it is so constructed that the pressure on its top is just enough in excess of the pressure under it to insure the valve remaining on its seat under normal conditions. A *piston*-valve is balanced by allowing the steam pressure to reach all parts of the central

cylindrical form, which moves in a cylindrical case, and connects the heads at each end.

BELL-CRANK.—Is a rectangular lever by which the direction of motion is changed through an angle, and by which its velocity-ratio and range may be altered at pleasure by making the arms of different lengths.

BRIDGE.—(See Valve Bridge.)

BUSHING.—(See Valve Bushing.)

CENTER.—That point within a circle which is equally distant from every point of the circumference.

Dead-) Is that position of the arms of a link-motion in which the center-lines coincide with the line of effort; that is, when the links are in the same straight line. Thus, when the crank, connecting-rod, and piston-rod are in a straight line, the engine is said to be on its (front or back) *dead center.*

Line of-) A straight line joining the centers of two wheels in gear. The dead line; that line in which a crank and connecting-rod stand when their axes form a straight line.

CLEARANCE.—Is the entire space beneath the valve-face which is filled with steam at the completion or beginning of a stroke, including the space between the piston and cylinder-head and the volume of the steam channels to the valve seat.

Exhaust-) Is the distance the exhaust edges of the valve lack of touching the exhaust edges of the admission ports when the valve stands central on its seat. It is sometimes referred to as *exhaust-lead.*

Inside-) In a *slide*-valve is not a portion of the valve proper, but the term represents the space between the inner or exhaust edge of the valve and the inside edge of the port opening when the valve stands midway in its travel. See Fig. 5.

Outside-) In a *piston*-valve, is the distance from the outside edge of the piston ring to the outer edge of the exhaust port when the valve stands central on its seat.

Piston-) The distance between the piston and the cylinder head at the end of a stroke.

COMBINATION-LEVER.—Is a lever which, indirectly, connects the valve rod to the crosshead and removes the lap of the valve for the admission of steam, by opening the port the desired amount of lead with the beginning of the stroke of the piston, thereby eliminating the effect of the angularity of the main rod. It is often referred to as the *lap and lead lever,* and is shown in Figs. 75 and 100.

COMPRESSION.—Is the term applied to the arrangement by which the exhaust opening is closed before the stroke of the piston is ended.

Point of-) The point at which compression begins is reached when the inside, or exhaust edge of the slide-valve reaches the inner edge of the steam port and completely closes the steam port and cuts off the exhaust steam. In other words, compression begins where the back pressure ends. The point of compression with a slide valve is shown by the arrow *a* when the valve is traveling in the direction of the large arrow A in Fig. 7, and is shown by the arrow *a*, in Fig. 55, with the piston valve.

CONNECTING-ROD.—A link which connects a rotating crank with a reciprocating rod and converts the reciprocating motion into a circular motion.

CRANK.—A crank is a device by which the reciprocating motion is converted into circular motion, or vice versa, consisting of a crank-arm, one end of which is fastened rigidly at right angles to the rotating shaft or axle, while the other end bears a crank-pin, projecting from it at right angles and parallel to the shaft. When the reciprocating part of a machine, as the piston and cross-head, is linked to this crank by a crank-rod or connecting-rod, one end of which works on the crank-pin and the other on a pin at the end of the reciprocating part, the to-and-fro motion of the latter imparts a circular motion to the shaft, and vice versa.

-*Arm.*) Is that part of a crank which joins the center of the shaft to be turned by it to the handle or pin to which the power of resistance is applied. See Fig. 15.

-*Circle.*) The circle described by a crank-pin; specifically, the circle in a valve-diagram, or the elevation, of a steam-engine, which represents the path of the crank.

-*End.*) The crank end of the cylinder is the end nearest to the crank shaft, and the farthest from the crank is the head end.

Leading-) Is the crank which leads when the engine is running ahead. The one which is less than half a revolution in advance of its fellow pin when the engine is running ahead.

-*Motion.*) The motion of a body driving or when driven by a crank. Such motion is called harmonic, as the reciprocating part tranverses the diameter of the circle in the same time in which the crank-pin is moving uniformly over the semi-circumference of the same circle.

-*Throw.*) May be defined as the distance from the center of the shaft to the center of the crank-pin.

-*Web.*) Is that part of a crank which connects the crank-pin to the shaft-hub. In short cranks this may be as large in cross-motion as either hub, while in long cranks it is usually cut away to make it as light as possible.

-*Wrist.*) The short length which forms the crank-pin surface to which the connecting-rod is attached.

CROSSHEAD.—Is a bar at the end of the piston-rod of an engine which slides on ways, or guides, fixed at the bed of the frame of the engine, which connect the piston-rod with the connecting-rod, or with a sliding journal-box in the crosshead itself.

-*Guides.*) Are the parallel bars between which the crosshead moves in a right line with the cylinder. They are sometimes called *motion-bars.*

CUT-OFF.—Is the cutting off, or closing, of the admission of live steam before the piston has completed its stroke, which permits the engine to utilize the expansive force of the imprisoned steam in the cylinder, behind the piston, after cut-off has occurred.

Point of-) The point of cut-off is the point of the piston traverse at which the valve closes to cut-off the admission of steam to the cylinder. It takes place when the valve arrives at the same position as for admission, but is moving in the opposite direction.

Short-) The cut-off is said to be *short* when the expansion is long. The expansion, or degree of expansion, is the reciprocal of the cut-off; for instance, if the cut-off is 3/5 of the stroke, the expansion is 5/3, or the steam has 5/3 the volume which is occupied when the valve is closed.

-Valve.) (See Expansion Valve.)

DEAD-CENTER.—(See Center.)

ECCENTRIC.—Is a circular disk, keyed to the axle in such a manner that the center of the disk and the center of the shaft do not coincide, thus imparting a rotary into a reciprocating motion. It is essentially an enlarged crank whose throw equals the amount of the eccentricity of its sheave, and is only used to avoid the necessity of cutting, or dividing, the crank shaft. See Fig. 14.

Adjustable-) One which is so constructed that the distance between the center of figure and the center of motion can be varied in order to vary the throw of the rod. This result is secured either by slotting the disk of the eccentric, or by mounting one eccentric upon another so that the effective eccentricity may be the sum of the difference of the eccentricity of each link, or may have intermediate values.

Angular advance of-) Is the angle between the center line of the crank of the engine and that joining the center of motion and center figure of the eccentric. See Fig. 17.

-Arm.) Is the arm used with outside valve gears in place of the eccentric employed with inside valve gears.

Backward-) The eccentric which is used when a locomotive is moved backward and the valve gear is adjusted accordingly.

Exhaust-) Is a term which assumes that the eccentric is fixed on the shaft, so that a constant valve travel, and fixed points of release and exhaust closure, may be obtained.

-Gear.) A term including all the links and other parts which transmit the motion of an eccentric.

-Rod.) Is the main connecting-link by which the motion of an eccentric is transmitted to the valve.

-Sheave.) May be defined as the inner portion of the eccentric, which rotates with the shaft. Marked AA in Fig. 14.

-Strap.) Is the band of metal which embraces the circumference of an eccentric, and within which it revolves. The eccentric-rod is attached to it. It is also called the *eccentric-hoop*, and is marked BB in Fig. 14.

-Throw.) Is the distance from the center of the shaft to the center of the eccentric disk. When the slide valve and the eccentric are connected by a rod the throw is equal to one-half the travel of the valve, or the lap plus the amount of opening of the steam port for the admission of steam. The distance between the letters C D in Fig 14.

ECCENTRICITY.—Of an eccentric, is the distance between the center of the shaft and the center of the eccentric, which is shown by the distance between the letters C and D in Fig. 14.

EXHAUST.—Is the release, or discharge, of expanded steam from the cylinder.

-*Cavity.*) Is designed to allow the escape of steam to the atmosphere, and, in the slide-valve, may be designated as the space between the exhaust edges. With the piston valve, however, the exhaust steam passes through the steam chest, around the outer ends of the valve, to the atmosphere.

-*Clearance.*) (See Clearance.)

-*Lap.*) The amount by which the edges of the working face of the valve project over the exhaust edges of the ports when the valve stands in its central position, symmetrical with the ports. See Fig. 4.

-*Lead.*) The amount by which the exhaust-port is open for the exhaust of steam, before the end of the pressure-stroke, or the commencement of the return stroke.

-*Nozzle.*) Is the blast-nozzle, or -orifice, which discharges the exhaust-steam into the stack to make a forced draft.

-*Port.*) Is the opening, or cavity, in the valve seat in which the exhaust passage terminates. See Fig. 2.

Point of-) That point in the stroke of the piston at which the valve opens to release from the cylinder the steam which has performed its work in driving the piston.

EXPANSION.—Is the increase in volume with corresponding reduction in pressure which occurs in a cylinder when the steam supply is cut off between the boiler and the cylinder, while the piston continues its motion.

Initial-) The expansion of water into steam which occurs in a boiler when the valve is opened to supply steam to the engine-cylinder. It is an amount sufficient to fill the cylinder and its clearance-volume up to the point at which cut-off takes place.

-Valve.) A valve, or slide, on the back of the main slide-valve for cutting off the steam earlier and more sharply than is done by the main slide-valve, thus causing a greater degree of expansive work of the steam than could be secured by the use of the main slide-valve alone.

Variable-) A term employed to designate expansion which varies under different conditions. It may be secured by changing the point of cut-off at will while the engine is at work, it may be fixed or secured at some pre-determined point of the stroke, or it may be automatic or self-varying.

GEAR.—Those parts of a machine which are concerned in effecting motion, as, for example, the parts of a locomotive from the cylinder to the wheels inclusive.

Full-) Such an arrangement of the valve gear mechanism as will give the longest period for the admission of steam.

Full-backward-) With the valve-gearing adjusted to produce backward motion of the engine.

Full-forward-) With the valve-gearing adjusted to produce forward motion of the engine.

LAP.—The amount that the valve extends over the steam port when the valve is in its central, or mid-position, or the space traveled by a valve after closing the steam port to or from the cylinder before reaching its central, or mid-position.

Amount of-) The lap is always measured at one end (not both) of the valve when it is in mid-position.

Effect of-) Is the cutting off of the admission of steam before the piston reaches the end of its stroke.

Exhaust-) Is that part of the valve which overlaps the steam ports when the valve stands in its central, or mid-position. It is sometimes referred to as *outside* and *inside* lap, and, occasionally termed *positive* exhaust lap because it represents metal added to the elementary valve, but the term *exhaust* lap is more pref-

erable, because it refers to the *outside* when a *piston*-valve is used, and the *inside* for a *slide*-valve.

Inside-) The inside lap of a *slide*-valve is the amount by which each of the exhaust edges of the valve overlap the inside edges of the steam port when the valve is in its central, or mid-position. It is also called *exhaust-lap,* or *cover,* and is shown by the space between the lines D and E in Fig. 4.

Negative-) The amount by which the exhaust edge of the *slide*-valve, when it is in its central, or mid-position, falls short of reaching the inner edges of the steam ports; the uncovered part is termed *negative-exhaust lap, negative-lap* or *inside-clearance,* and is shown by the letters C and C in Fig. 5.

Outside-) Is that part of the valve flange which overlaps the steam port, when the valve stands central upon its seat. It is also called *steam-lap,* or simply *lap,* and is shown by the space between the letters A and B in Fig. 4.

Seal-) (See Seal.)

Steam-) The distance the valve is moved from its central position to permit *admission* or *cut-off* to occur. See *outside-lap.*

LAP & LEAD LEVER.—(See Combination Lever.)

LEAD.—The distance, or width of port opening, for the admission of steam when the crank is on dead center.

-Angle.) Is the angular displacement of the center line of the eccentric ahead of its normal relation to the center line of the crank, which is given so that the opening of the port may precede or lead the beginning of the piston stroke by a predetermined amount. The point is shown in Fig. 17.

Constant-) Implies a uniform lead, which does not change for varying grades of expansion.

Equal-) Means an equality of, or the same amount of lead at both ends of the cylinder.

Exhaust-) The lead on the exhaust side of the steam port. (See Exhaust-lead.)

Negative-) A term used to designate the distance which a valve has to travel, at the beginning of the stroke, before it opens the port. Or the angle through which the crank has turned from the dead center when the valve opens to admit steam. It indicates the amount the *steam-edge* of a valve overlaps the steam-port *at the commencement* of the stroke.

Positive-) The amount, or distance, the valve has opened the port before the piston commences its stroke.

Steam-) The lead on the steam-side, or at the admission port.

Variable-) Is the opposite of constant-lead. It is lead which varies, or is liable to change, for varying grades of expansion.

LINEAR-ADVANCE.—The distance the valve is moved from its central, or mid-position when the piston reaches the end of its stroke. Or the distance by which the port is open when the piston begins its stroke.

LINK.—Is a bar in which there is a slot, and the latter may be straight or may be a curve of large radius. The adjustment of the link block in the link determines the motion or direction of the locomotive, as well as the travel of the valve.

-Block.) The block attached to the valve-stem, actuated by the link-motion.

-Carrier.) A metal piece which carries the link in a locomotive valve gear; a *stirrup.*

-Foot.) Is the extension of the link used in the Walschaert gear to which the eccentric rod is connected.

-Hanger.) Is the suspension-rod fastened to the saddle-pin in the Stephenson link-motion valve gear, by which the weight of the link and of the eccentric-rods is borne, and by which the link is raised and lowered.

Main-) The link that connects the end of the radius bar on a steam-engine to the valve-stem.

-Motion.) A system of gearing for controlling the valves, for the purpose of starting or reversing the engine, and for controlling the cut-off.

Oscillating-) (See Oscillating.)

Radius of the-) Is the distance from the center of the main shaft to the center of the rocker box, minus any backset given the rocker arm. It equals the exact length of the eccentric blades, plus the distance from the link pin arc to the link arc. On outside radial gears, it is the length of the radius rod, or the distance from the valve-stem connection to the center of the link carrier.

-Saddle.) The plate, or bar, bolted to the side of the link, and to which the link-hanger is attached.

-Saddle-stud.) The pin, or extension, on the saddle, to hold the link hanger in position.

Slip of the-) Is the distance that the link-block moves up and down in the link while in motion, without any movement of the reverse lever.

-Slot.) The slot, or opening, in the curved member of a link-motion for a locomotive and other reversing engines. This member is called the link, and in the opening, or slot, slides the link-block by which the valve is operated, as the two eccentrics operate the link itself.

-Stirrup.) (See Link-carrier.)

-Swing.) The greatest distance that the bottom of the link will swing from front to back in full gear.

Union-) A bar or link which connects the lower end of the combination lever with the crosshead; also called the lap and lead connector. See Fig. 102.

LINKING-UP.—The act or process of altering the position of the links or link-blocks in a reversing and cut-off engine, to produce an earlier cut-off and greater expansion. Also called *hooking-up.*

LOST-MOTION.—Any difference of motion between the driving parts, or between the parts of a locomotive, or any machine.

MAIN-*pin.*—The crank pin to which is connected the back end of the main rod.

-*Rod.*) The rod that connects the crosshead to the main pin.

MOTION.—Is the movement from one point or position in space to another.

Center line of-) By "the center line of motion of the valve gear," is meant a line drawn through the center of the shaft parallel to the direction in which the valve moves, when no rocker, or other mechanism between the shaft and valve, is used.

Direct-) A valve gear is said to be a direct motion one when any movement of the eccentric or eccentric rod causes the valve to move in the same direction. A forward movement of the eccentric will thus push the valve forward.

Indirect-) A valve is said to be indirect when a forward movement of the eccentric causes the valve to move back.

Lateral-) Is the end-play, or freedom of movement, of an axle in its boxes.

Link-) (See Link-motion.)

Lost-) (See Lost-motion.)

Reciprocating-) A motion of a part out and back over the same path, first in one direction and then in the other; also the parts which have such back and forth motion.

Reverse-) (See Reverse-motion.)

-Rod.) The rod which communicates motion from a rock-shaft, driven by the eccentric, to the valve-stem. Also called a *transmission-bar.*

Valve-) Is the motion transmitted to the valve by the valve gear. In other words it is the travel of the valve.

OVER-TRAVEL.—The distance a valve travels beyond the point necessary to give a full steam port opening. See Fig. 8.

OSCILLATING.—Anything which moves backward and forward, or swings to and fro; a link on a block.

PISTON-*Area.*) The area of the flat surface or end of the piston; the area of the cross-section of the bore of the cylinder.

-*Clearance.*) (See Clearance.)

-*Displacement.*) The distance the piston has traveled from the end of its stroke. The volume of a cylinder having the same diameter as the piston and a length equal to its stroke.

-*Head.*) The disk which is fitted closely to the interior of the cylinder, and is the direct receiver or transmitter of the power developed; distinguished from the *piston-rod.*

-*Rod.*) The rod that connects the piston head to the crosshead, which, in turn, is connected to the main pin by the main rod.

-*Stroke.*) A stroke of the piston is equal to the diameter or the circle described by the crank-pin. The distance of its travel from one end of the cylinder to the other.

-*Travel.*) (See Piston-stroke.)

-*Valve.*) Is a slide valve of cylinder form, or a spool shaped device, which moves in the direction of its axis. It is hollowed in its center and fitted with packing rings in suitable grooves at each end, to insure a steam tight joint between the live and exhaust steam passages.

PORT-*Opening.*) The extreme uncovered distance of the port between the steam edge of the valve and the steam edge of the port. See Fig. 2.

PRE-ADMISSION.—(See Admission.)

PRESSURE.—*Absolute-*) Is the pressure reckoned from a vacuum; the pressure shown by the steam gauge, plus the pressure of the atmosphere.

Boiler-) The pressure above atmospheric pressure; the pressure in a boiler shown by a steam guage.

Back-) Is the pressure on the exhaust side of the piston, caused by the steam left in the cylinder after the exhaust opens, which opposes the advancing motion of the piston during its return stroke, and causes a reduction of the work to be done by the piston through the action of live steam.

Effective-) The unbalanced pressure on a piston; the net pressure available for doing work; the absolute pressure on the pressure side of the piston minus the back pressure on the exhaust side of the piston; usually called *mean effective pressure.*

High-) Formerly, a phrase denoting all steam engines working at pressure materially higher than atmospheric pressure, but now a relative term.

Initial-) The pressure in the cylinder at the beginning of the forward stroke.

Low-) The precise signification of the term is undetermined, but the standard of pressure is steadily rising, so that engines that were formerly considered high-pressure are now looked upon as low-pressure engines.

Mean-) The average pressure in the cylinder from the beginning to the end of a stroke. See Effective-pressure.

-Plate.) A cover-plate attached to the inner cover of the steam chest, designed to relieve a slide valve of friction by preventing the steam pressure from coming into contact with a portion of the upper surface, or top, of the sliding valve.

Terminal-) The pressure of the steam in the cylinder at the end of the stroke, or the pressure it would have if the exhaust port was not opened until the end of the stroke.

QUADRANT.—A link, with a slot struck with an arc of a circle as center-line, used in the operation of valve gears, in which the lock or latch of the reverse lever can be engaged to hold the lever in the desired position. Called also a *valve-sector.*

RADIUS-BAR.—One of a pair of rods pivoted at one end and connected at the other with some concentrically moving part which is necessary to keep at a definite distance from the pivot or center. Also called a *radius-rod.*

-Link.) (See Link-radius.)

REACH-ROD.—A rod which connects the reverse lever in the cab to the bell crank on the reverse shaft of the valve gear.

RECIPROCATING-MOTION.—(See Motion.)

RELEASE.—The opening of a port for the escape of steam which has been confined behind the piston to drive it through its working stroke.

Point of-) The place in the stroke of the piston at which the exhaust valve opens, releasing the steam which has been confined behind the piston. See Fig. 7.

REVERSE.—To cause to act in a contrary direction; to give an exactly opposite motion or action to the crank, or to that part to which the piston-rod is attached.

-Lever.) A lever which operates the valve gear and the valve so as to reverse the action of the steam on the piston and thus change the direction of motion.

-Link.) The link or guide in which the link-block of a reversing valve gear slides.

-Motion.) Any mechanism for changing the direction of an engine. A common device used is a loose eccentric working between straps on the engine-shaft, so that it may be made at will to assume an angular relation to the engine crank, which shall be correct for forward or for backward motion.

-Shaft.) A shaft connected with the valves of an engine in such a manner as to permit a reversal of the direction of the motion of the engine-shaft.

-Valve.) The valve of a reversing-cylinder. It may be a plain, slide or reversing type.

-*Yoke.*) A part used in the Baker valve gear to displace the link, and to reverse the motion of the engine.

ROCKER-ARM.—An arm or lever attached to a rock-shaft so that it may receive or give a reciprocating angular motion. It changes the motion of the Stephenson gear from direct to indirect.

-*Bar.*) A bar or lever mounted on a shaft which has an oscillating motion.

-*Box.*) The box or bearing for a rock-shaft.

SADDLE-PIN.—The pin which is fastened to the U shaped forging, and on which the link is suspended by its hanger.

-*Stud.*) (See Saddle-pin.)

SEAL.—The amount a valve overlaps the edge of a port to seal it, and prevent leakage, when the valve is in a closed position for the particular port.

STUFFING-BOX.—A box cast around the hole through which the piston rod passes, in which is laid, around the rod, and in contact with it, a quantity of hemp or metallic packing. The packing is lubricated with oily matter, and a gland or flanged cylinder is placed on the top of it which is pressed down by screws, so as to squeeze the packing into every crevice. It is also called the *packing-box.*

TRANSMISSION-BAR.—A bar of steel or iron used to connect the link block to the rocker arm, or valve rod

connection, on an engine where the rocker is placed ahead of the link. Also called a *motion-rod.*

TRAVEL.—(See Valve-travel.)

TUMBLING-SHAFT.—The tranverse shaft in a Stephenson reversing link-motion from arms of which the link proper is suspended or controlled in position. When the engine is horizontal, the arms have a rocking or tumbling motion through the necessary angle to move the link from the position from full forward gear to full backward gear.

VALVE.—

Admission-) (See Admission-valve.)

Balanced-) (See Balanced-valve.)

-Block.) A moving block, such as a link-block or cross-head, used to connect two moving parts.

-Bridge.) The wall, or partition, which separates the steam-ports from the exhaust cavity. See Fig. 2.

-Bushing.) A cylindrical lining, or form, pressed into the steam chest, in which the piston valve operates.

-Chest.) The box, or casing, inclosing the valve.

-Clearance.) (See Clearance.)

Cut-off-) (See Cut-off.)

-Displacement.) The amount a valve is moved to the right or left from its mid-position.

-*Face.*) The finished surface of the valve, on which it slides, or moves, and fits to open and close the passages it is to control.

-*Gear.*) A collective term used to designate the entire mechanism, composed of the parts from the cylinder to the wheels, employed to communicate motion from the wheels to the valve.

-*Line and line.*) A valve is said to be set *line-and-line* when it has no lead in full forward gear. That is, when the line of the edge of the valve coincides with the line of the edge of the steam port at the beginning of the stroke. See Fig. 1.

Mid-position of-) The position of the valve when the eccentric is perpendicular to the line of stroke; the middle of the travel when a valve covers both steam ports the same amount.

-*Motion.*) (See Motion.)

-*Over-travel.*) (See Over-travel.)

Piston-) (See Piston-valve.)

-*Plate.*) A flat plate forming part of the seat of a valve, which can be renewed upon wearing, or in case of leakage, without discarding or refinishing the entire casting.

-*Port.*) One of the openings which terminate in the valve-seat, by which the passageways to the cylinder are placed in communication with the steam pressure;

it also serves as a terminal for the discharge of exhaust steam from the cylinder. See Fig. 2.

-*Port-opening.*) Is the *uncovered* part of the port, and it is generally less than the steam-port width.

-*Rod.*) The rod which communicates motion from the rockshaft, driven by the eccentric, to the valve-stem.

-*Seal.*) (See Seal.)

-*Seat.*) The surface upon which the valve rests and slides. See Fig. 2.

-*Sector.*) (See Quadrant.)

Slide-) A slide-valve is one which slides to and fro, over and upon its seat, partially or wholly opening and closing a port or ports formed in the valve seat, which are to be alternately opened and closed for the admission and exhaust of steam to and from the cylinder.

-*Stem.*) A rod, similar to a piston-rod, by which a valve is moved.

-*Stem-guide.*) A hollow bar, or stud of iron, in which the valve stem travels, placed in line with the steam chest to insure a straight travel of the valve stem.

-*Spindle.*) The stem, or axis, on which a valve is fastened, and which transmits motion from outside of the steam, or valve, chest to the valve proper, within it. The spindle, or stem, slides in and out through a stuffing-box.

-*Strip.*) A straight piece of metal acting as the valve-ring and used on rectangular balanced valves instead of a ring, to serve as a packing-strip between the valve and the pressure-plate.

-*Travel.*) The linear distance the valve travels in moving from one extreme position to the other. Twice the eccentricity, or throw of the eccentric.

-*Yoke.*) The loop of metal, or band, around a slide-valve, which connects the valve to the valve stem.

WIREDRAW.—Wiredrawing means reducing the area through which the steam may flow, thus materially reducing its pressure after passage through the constricted opening.

Assuming now that the reader has studied, and is familiar with, the various terms defined, we shall proceed to describe the construction and application of the valve, and, later on, the different types of modern valve gear.

THE SLIDE VALVE.

There are but two general types of slide-valve—the *flat* valve and the *circular piston* valve, but each type is represented by many different styles, which may be divided into two classes: *direct* and *indirect* valves. A valve is said to be *direct* when it moves to the left and opens the right-hand port, and closes the same when moving to the right, and is classed as *indirect* when it moves to the right to open the right-hand port and travels to the left to close it. The flat *D* slide-valve is an example of the direct type, while the piston valve is usually of the type classed as indirect.

The oldest, and simplest, form now in use is the flat *D* shaped slide-valve, so it will be considered first.

Invention.

A crude form of the slide-valve, which is shown in Fig. 1, was first used near the end of the eighteenth century, and Mr. Matthew Murray, of Leeds, England, is generally accredited as the inventor. The form of this valve was later improved upon by Mr. James Watt,

but the invention of the long *D* slide-valve, which, in a modified form, is used at present, is a product of one Murdock, who was an assistant of Watt.

The plain D slide-valve was simple of construction and durable, but the efficiency of some of its modifica-

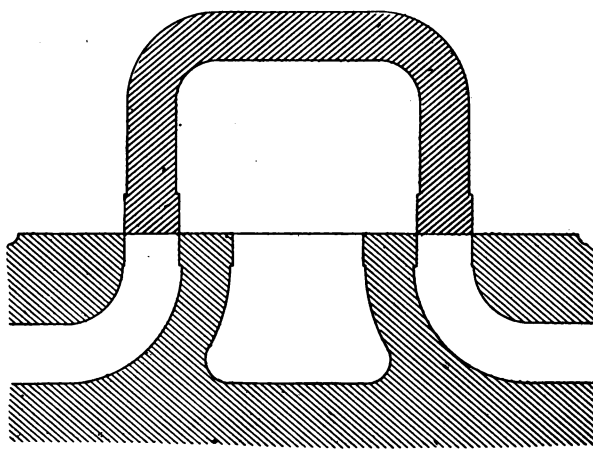


FIG. 1.

tions, as, for instance, the balanced slide valve, has been questioned, and it has formed the basis of severe, if not continuous, criticism, but it still occupies a prominent place in modern locomotive construction.

Functions.

The efficient and economical operation of a steam valve requires that a certain amount of steam, ample

for the load to be moved, must be allowed to enter and escape from the cylinder at fixed predetermined points of the piston travel; for example:

First. Steam must not be admitted at both ends of the cylinder at the same time;

Second. The exhaust of steam from the cylinder should commence a little before, but never later than, the admission of steam at the opposite end, and

Third. The outer edges of the steam ports should be covered completely when the valve stands in its central position, so as to prevent steam from passing from the steam chest into the exhaust port.

A valve which will not fulfill these requirements cannot be used with success or economy.

Satisfactory results can only be obtained from a properly constructed valve, and its size is governed solely by the proportions of the valve seat, on which it slides. For this reason, to assist the reader in understanding the design of the valves now in use, we present a general outline of the valve seat.

Construction of the Valve Seat.

The valve seat is, generally, cast on top of the cylinder, and it must be planed perfectly smooth, to avoid as much friction as possible, and to insure a steam tight fit. It is provided with two steam channels, or ports—one in each end, which also serve as exhaust passages;

and one exhaust cavity, or port, situated in the center between the two end ports, all of which terminate in the smooth flat surface called the valve seat, or base.

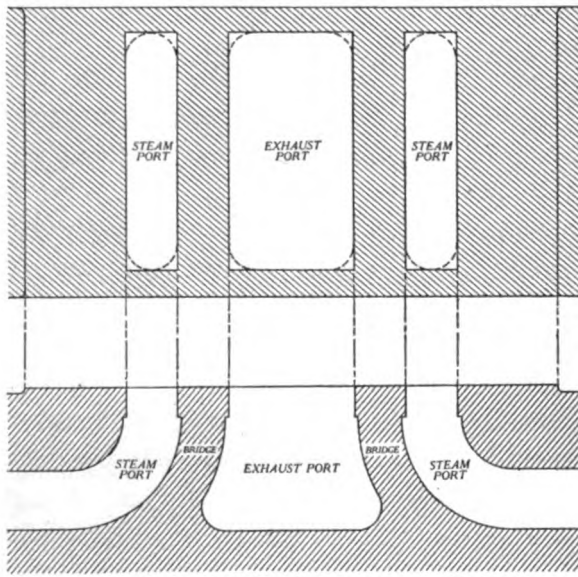


FIG. 2.

Fig. 2 shows a top view, and a cross-section, of the valve seat. The steam ports communicate with the boiler and the exhaust cavity and port, alternately, while the exhaust cavity and port terminate with the atmosphere, through the exhaust nozzle and stack, for the discharge of steam which has been used.

To insure good service, both the valve and the valve seat should be made of hard cast iron. It occasionally happens, however, that the cylinder casting is made of softer metal than it should be, and, as a result, the valve seat wears rapidly. In such cases a false seat must be applied, just as is done when the valve seat is faced down to its limit. The practice of substituting false seats should be avoided when possible, for they are expensive, and are almost sure to give trouble unless the work is performed in the best possible manner.

The valve seat should always be raised a little, from 1½ to 1¾ of an inch, above the surrounding metal of the cylinder and should be a trifle shorter than the extreme travel of the valve, to prevent the wearing of a shoulder on the valve seat.

The longitudinal width of the valve seat is not so important, except where special valves of the Allen type are used, but when possible it should be made wide enough to allow a surface for the valve equivalent to the width of one bridge when the valve is at extreme travel position, as hereafter shown in Fig. 8, unless such allowance would result in the wearing of a shoulder on the valve seat when the engine is hooked-up in the working notch.

Bridges.

The bridges should be made as thin as possible, to reduce the pressure required to move the valve, but of sufficient strength to resist the highest pressure. The thickness of the cylinder walls is generally considered a safe measurement for that of the bridges. They are, however, usually made a little wider, but the face may be beveled without materially affecting its strength; however, it should be remembered that a reduction of 1/8 inch in its width will reduce the width of the valve 1/4 inch, and result in a decrease of the area on top. Over-travel should also be considered, and sufficient surface be provided for the valve when it is in extreme travel position to secure a steam tight joint, about 1/4 inch being sufficient. The wear must also be taken into consideration, for too narrow a bridge would not maintain a steam tight joint.

As a general rule the width of the bridge is less than the width of the steam port, and on American locomotives it varies from 15/16 to 1 1/4 inches.

Port Opening.

An important task in designing a valve is to provide proper port areas for the admission and exhaust of steam. If the cross-sectional areas of the port are too

small, the steam will be choked, or throttled, its velocity increased, and its pressure reduced. This result is generally referred to as *wire drawing,* the effect of which will be considered later on. On the other hand, if too large a channel be provided the clearance volume will be unnecessarily increased, and economy of the engine reduced proportionally.

The proper size is that which will insure the largest port opening practicable at the running cut-offs, while reducing the preadmission in full gear to a negligible quantity. Any port opening desired at the working cut-off may be secured by giving the necessary lead to the valves. But to reduce preadmission, and render it negligible, so far as starting is concerned, lap should be added. The more lap given a valve the less faulty steam distribution is to be expected, and vice versa.

Even though changes in speed may not affect the extent of port opening, the length of time of its opening will be reduced as the speed is increased, and, as a result, a smaller amount of steam will be admitted per stroke, depending, of course, upon the amount the speed is increased.

In some cases the throttling of steam is unavoidable, but it can be diminished by a quick opening and closing of the valve.

Location of Port Openings.

It is advisable, when possible, to use a small valve, and place the ports as close together as practicable, for friction and weight are important considerations, and considerable work is required to accelerate and retard the moving masses employed; besides, the wear and tear of the valve gear will be in proportion to the acting forces. The unbalanced *D* slide-valve, as will be hereafter shown, has steam pressure over its entire back, or top, and the friction and weight are necessarily increased in proportion to its size.

Steam Ports.

No general rule governing the size of the steam ports can be stated, for considerable difference of opinion exists in regard to this, and it has not been determined with any degree of accuracy. But it may be said the size depends, to a considerable extent, upon the speed of the piston and the dimensions of the cylinder.

When the admission of full boiler pressure steam to the cylinder is desired, for high-speed engines, large ports are necessary, to secure the free admission and release, but small ports are more satisfactory and preferable, when they can be used, for they keep the valve motion within practical limits.

The admission of live steam does not require as large a port opening as is necessary for the exhaust of the expended steam, and if the port is large enough for the exhaust to pass out without appreciable back pressure it will be ample for the admission of steam. In high-speed engines there is seldom any difficulty experienced in getting steam into the cylinder, but occasionally trouble is encountered in getting the steam out of the cylinder fast enough.

It has been demonstrated that for a piston speed of 600 feet per minute, a good exhaust will be secured when the area of the steam port is 1/10 the area of the piston, if the steam is in an ordinary state of dryness. Of course, for slower piston speeds the steam port area will be less, and will be proportionally larger for increased piston speed; in fact the rule of proportion may be stated as follows:

Given piston speed in feet per minute $\times$ .1 $\div$ 600 = the port area in fractional parts of the piston area.

In other words, the specified piston speed in feet per minute multiplied by 1/10, and divided by 600, will equal the port area in fractional parts of the piston area.

Now, if we multiply the area in square inches of the piston by the port area, in terms of the piston area, we can ascertain the number of square inches that the steam port must contain; for example:

To find the steam port area for a cylinder 24 inches in diameter, and a piston speed of 650 feet per minute, we may put our data in the following form:

$$\frac{650 \times .1}{600} = .108$$

that is, the port area must be equal to 108/1000 part of the piston area. The area of a piston 24 inches in diameter is 452.39 square inches, hence $452.39 \times .108 = 48.85812$. This means that the steam port must be 48.85 square inches for this particular piston speed. Of course, for a slower piston speed, this port area should be less, for instance, if the piston speed is to be 500 feet per minute, and the diameter of the cylinder 24 inches, as in the previous example, we have—

$$\frac{500 \times .1}{600} = .083$$

and the piston area $452.39 \times .083 = 37.4837$ square inches for port area.

To ascertain the proper area for a steam port it is only necessary to multiply the area of the piston in square inches by the number opposite the piston speed shown in the following table, which has been found to give good results:

Speed of piston in feet per minute.	*Multiplied by*
100	.016
150	.025
200	.033
250	.041
300	.050
350	.058
400	.066
450	.075
500	.083
550	.091
600	.100
650	.108
700	.116
750	.125
800	.133
850	.141
900	.150
950	.158
1000	.166

Another rule for ascertaining the area of the steam port is this; multiply the square of the diameter of the cylinder by .078.

The length of the steam port is usually made equal to, but should never be less than three-fourths of, the

diameter of the cylinder. Better results are obtained from a long port, as it increases the opening for admission and release, reduces the travel necessary to obtain a full port opening, and reduces the area on the back of the valve, so that the valve can be moved with less power.

Assuming that the length of a steam port, with square ends, has been decided upon, the necessary width of the port can be ascertained by dividing the area of the steam port by its length. If predetermined width is specified, divide the area of the port with the width, and the quotient will be the length of the port.

It is a general rule, in practice, to make the port opening coincide with the passage, or channel. It is also considered good practice to have the ends of the steam ports form a semi-circle, as shown by the dotted line at the end of the ports in Fig. 2, for ports with square ends are liable to wear grooves and ridges, although they admit and cut off steam along the entire edge of the part at the same moment. Another advantage derived from constructing the ends of the ports semi-circular is the additional strength given to the bridges.

Exhaust Port.

The exhaust port should be made more than twice the width of the steam port, especially with over-travel,

to secure the free escape of steam and reduce the back pressure as far as practicable; if made smaller it would throttle or choke the valve, as shown by Fig. 8. On the other hand, it should not be made too wide, or it would unnecessarily add to the size of the valve and increase the pressure on it, thereby increasing friction and the wear and tear on the valve gear. In addition, it may be noted here that the size of the exhaust port has no material influence on the valve. With unbalanced valves, however, it is considered good practice to reduce the exhaust cavity as much as possible; in such a case the exhaust cavity opening at the extreme end of the valve travel may be reduced to about five-eighths of the width of the steam port area, as a result of the valve overlapping.

The general rule for ascertaining the width of the exhaust port may be stated as follows: Add the width of one steam port to one-half the travel of the valve, and from that amount subtract the width of one bridge. Another rule for determining the area of the exhaust port, is to multiply the square of the diameter of the cylinder by .178.

Assuming now that the reader has a general idea of the form of the valve seat, the question arises: What size shall we make the valve?

Construction of the Valve.

We do not believe an analysis of the various theories advanced in favor of a large or small sized valve would

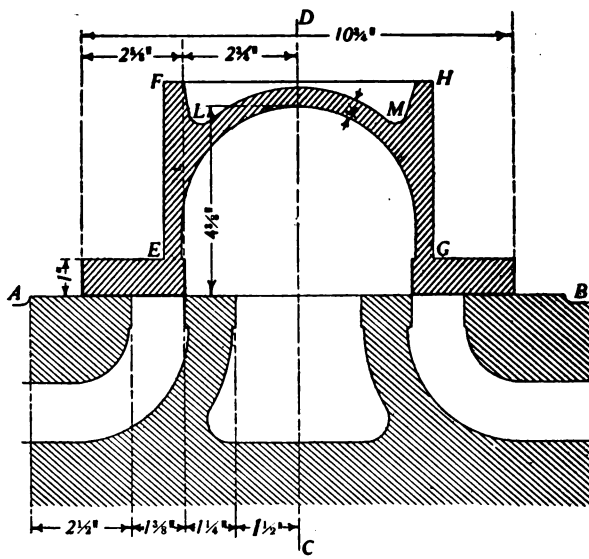


FIG. 3.

be of any benefit to the reader at this time; therefore, we shall proceed to construct a valve for the valve seat previously illustrated.

The face of the valve seat is here shown by the line A B in Fig. 3, and we will now draw the perpendicular line C D through the center of the line A B to locate the

center of the exhaust port and the center of the valve when it stands in mid-position.

If the valve be constructed without inside lap, or inside clearance, its inner edges must coincide with the inner edges of the steam ports, and its inside width must equal that of both bridges and the exhaust port, or be $1\frac{1}{4}+1\frac{1}{4}+3=5\frac{1}{2}$ inches.

Next we will draw the heavy base lines from the outer edges of the bridges to the outer edges of the steam ports, a distance of $1\frac{3}{8}$ inches.

This will give us a valve which is termed *line and line,* as shown in Fig. 1, but such a valve would not be satisfactory, for reasons we shall soon make clear; therefore, we must give the valve lap, and, without stopping at this time to state the rule for determining the proper amount of lap for any particular valve, we will add the width of one bridge, or $1\frac{1}{4}$ inches, at each end; so that our heavy base line is now $1\frac{1}{4}+1\frac{3}{8}=2\frac{5}{8}$ inches.

Now in determining the thickness of the metal necessary to support the valve face we must allow something for wear, and its depth should be enough to insure a proper support for the valve yoke, so we will make the flange twice the thickness of the sides or top of the valve, or say 1 inch.

Now, to lay out the exhaust cavity, we will draw two heavy lines 1 inch high, to correspond with the outer

edges of the valve flanges, from the inner edges of the steam ports, making the two lines perpendicular to the line A B. The depth of the exhaust cavity is sometimes made a trifle less, but it should not be greater, than the width of the exhaust port plus one steam port. We believe, however, the larger size will give better results so we will make it $3+1\frac{3}{8}=4\frac{3}{8}$ inches.

Now we must decide upon the height of the outer walls of the valve, and upon the shape of the top of the valve, for some valves are constructed with a round top, as illustrated in Fig. 1, while others are made with a square top.

If the flange of a valve face is allowed ½ inch for wear, then the thickness of the top and sides for either form of valve may be made the same, for they do not come in contact with a wearing surface, and are not subject to wear.

Without attempting to decide the advantages to be derived from the adoption of either form of top, or crown, we shall assume the round top is, for some reason, undesirable.

Now returning to Fig. 3, we will draw a light line, from the top of the outer end of the metal flange supporting the valve face, to a point ½ inch from the inner edge of the exhaust arch.

It is now obvious the outer sides designated E F and G H must extend upward a distance in excess of the

depth of the exhaust cavity (4⅜ inches), and the thickness of the exhaust arch (½ inch), or a total of 4⅞ inches to secure a square top, so we will draw the two outer lines to a point 5 inches from the valve face.

The two depressions, or recesses, indicated by the letters L M are solely for the purpose of saving metal and dispensing with unnecessary weight of the valve.

A flat top for the valve is preferable because the increased height of the valve provides a larger surface against which the valve yoke may bear, and it gives a form which may be laid on its back and secured to the planer without difficulty, when the valve face is to be planed. The flat top also furnishes a convenient base for the insertion of balance springs and strips when it is desired to convert a plain slide valve into a balanced valve.

Valves Without Lap.

By referring to Fig. 1 again the reader will observe that the valve is in a position described as *line and line;* that is, as previously explained, the inside and outside edges of the ports and the inside and outside edges of the valve coincide; by moving this valve in the slightest degree, either to the right or left, steam will be admitted in one port, and, at the same time, the opposite port will be opened to the exhaust of steam. Thus, the duration of steam admission at one end would be the same as

that of exhaust at the opposite end of the cylinder. A valve of this kind would admit steam to the cylinder during the whole stroke of the piston and be extremely wasteful, because it would not use the expansive power of the steam.

Such a valve is generally modified so the admission of steam will be cut off before the piston reaches the end of its stroke, so as to obtain the advantage of expansion.

It has also been demonstrated that, among other advantages, a considerable saving in coal can be obtained by permitting the admission of steam for only a part of the stroke and allowing the steam in the cylinder to expand for the remainder of the stroke.

Valves without lap are not in use at the present time, however, and the illustration is beneficial only in showing the development in valve construction.

Valves With Lap.

The valve was given lap in this country long before its advantages as an element of economy were fully appreciated here, or understood abroad, and there is authority for the statement that as early as 1829 lap was used on an engine designed to run a steam carriage in New York, and in 1832 it was used on an engine of a steamboat, because it was generally believed, even at that time, that lap provided for the expansion of steam.

It was soon recognized as an important improvement in locomotive construction, but the economy of fuel it brought about was not thoroughly understood. In fact, many eminent mechanical engineers of that time were of the opinion that the early opening of the exhaust simply reduced the back pressure, but that theory was exploded when the Indicator was invented and applied to the locomotive, for it was then demonstrated that additional work was derived from the steam by using it expansively.

The term lap, as previously explained, is used to designate the outside or steam edge of the plain slide-valve, or the inner edge of the steam or admission port when piston valves are used. Giving the valve lap consists of extending or lengthening the metal of the valve face so that the edges of the valve when in mid-position will extend a distance over the steam ports, as shown by the space between the letters A and B in Fig. 4. The extension is added so that the valve will close the steam port before the piston completes its stroke, and the point where the steam port is closed is called the point of cut-off; this point depends on the speed of the engine, etc.

On some engines the inside edges of the valve cavity are slightly cut away so that they do not reach the inner edges of the steam port, when the valve stands in mid-position. This space is termed inside clearance. The

great majority of locomotives have outside lap, but high-speed engines have little or no inside lap, but are usually given inside clearance.

The amount of lap to be given any engine depends, of course, upon the character of the service it will be required to give, and it varies from ½ to 1½ inches. Valves having considerable lap are less likely to give faulty steam

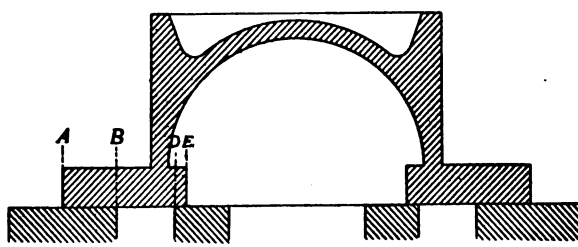


FIG. 4.

distribution than valves having short lap. With a given travel the period of expansion will be increased in proportion to the extension of the lap.

Inside Lap.

Inside lap delays the release of steam and prolongs the period of expansion; hastens compression and, as a result, increases it; tends to choke or retard the exhaust, but does not effect admission or the point of cut-off. Inside lap is indicated by the space between the letters D and E in Fig. 4.

No definite rule for determining the proper amount of inside lap can be given, for there is a great difference of opinion on the subject. The amount, however, is generally small, and depends upon the service the locomotive is required to give, varying from zero to ⅛ of an inch. In numerous locomotives, especially passenger locomotives operated on a level track, the valves have no inside lap, but for slow-speed engines, such as freight or switching locomotives, which are not required to ascend steep grades, a small amount of inside lap will be beneficial.

The old practice of giving valves 1/16 of an inch inside lap has been almost altogether discontinued, and in place of it the valves are made line and line, or given inside clearance.

Inside Clearance.

Inside clearance, which is sometimes called exhaust clearance, or negative inside lap, is given to a valve to reduce the resistance to the movement of the piston, and it should be sufficient to take full advantage of the nozzle opening, for it delays compression, but hastens the exhaust release, and, as a result, produces a more speedy engine. It opens the exhaust earlier, keeps it open longer, and, as a result, reduces the back pressure, but it has no effect on the point of admission, or cut-off.

The detrimental effect of inside clearance is the sacrifice of water and fuel for increased speed. From an

economical standpoint it is expensive, because it does not obtain the advantages of steam expansion, but allows the steam to be exhausted while at high pressure in the cylinder. This can be overcome, however, by giving an equal amount of inside clearance in the opposite ends of the cylinder in proportion to the exhaust edge of the valve lip.

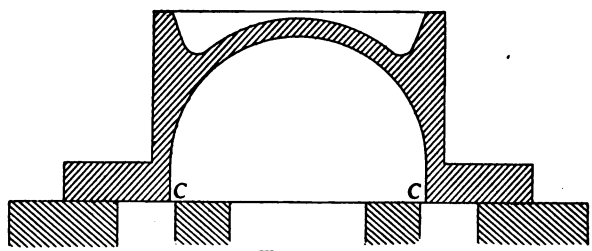


FIG. 5.

Of late years the amount of inside lap has been gradually reduced, and at present valves are sometimes given from $\frac{1}{8}$ to $3/16$ of an inch inside clearance on engines operating at high speeds, but there is considerable difference of opinion regarding the beneficial results obtained by the change.

Of course an increase of inside clearance will result in a diminution of efficiency at slow-speed, but it is beneficial on high-speed locomotives where long fast runs are the rule, or on engines with small wheels which run at a moderate rate of speed, for it causes the release of steam in the cylinder to occur earlier and, as a result,

compression later in the stroke. It all depends, however, upon the character of the service required of the locomotive, and it may be safely stated that ¼ of an inch inside clearance is excessive for ordinary conditions.

One of the early, if not principal, objections to giving a valve inside clearance was the fact that it placed the steam ports in communication with each other, as well as with the exhaust port, and it was thought the steam would blow through and be wasted. The extent of time, however, during which the steam ports are in communication depends partially on the amount of inside clearance, but principally upon the travel of the valve and the speed of the engine.

By referring to the spaces indicated by the letters C and C in the illustration, Fig. 5, it will be seen that the intercourse between the two steam ports cannot result in the anticipated loss of live steam, for both of the steam port channels are acting as exhaust ports while they are in communication, so the only detrimental effect which will develop is an interference between the exhaust steam from the two ends of the cylinder which meets in the exhaust cavity.

It may be added that the assumed detrimental effects of inside clearance are entirely overcome, or rendered

harmless, when the speed of the engine reaches, or exceeds, the rate of 50 miles per hour.

Admission.

The admission of steam behind the piston, to drive it through its stroke, commences when the valve opens the steam port for pre-admission, or lead. Admission continues until the valve reaches the end of its travel and then returns to the cut-off position, that is, to the position at which the supply of steam to the cylinder ceases. During this period of admission it is essential that sufficient steam be taken into the cylinder to move the piston for the work required during the entire stroke.

Cut-Off.

The cut-off of the admission of live steam is one of the most important events of the piston stroke. It does not take place at the same point on the forward and backward strokes, respectively, because the slanting, or obliquity, of the connecting rod to the line of stroke results in the piston being more advanced in the forward than in the backward stroke than it should be to correspond precisely with the position of the crank. This disparity will be explained later on.

The percentage of the piston's stroke at which cut-off occurs depends entirely upon the position of the reverse

lever in the quadrant. For example, if the stroke of the piston is 32 inches, economy of steam can be obtained by placing the reverse lever at a point which will cause the valve to cut-off at about 8 inches, which would be one-quarter of the piston stroke; and allowing the expansion of steam in the cylinder to supply the force for the remaining 24 inches of the piston stroke. If, how-

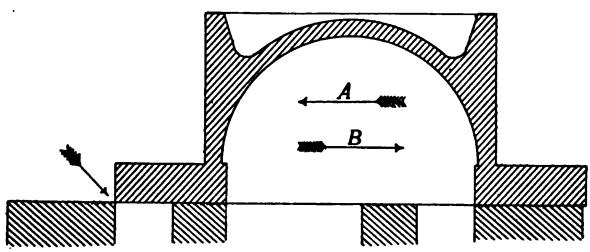


FIG. 6.

ever, the desired speed cannot be obtained by such an arrangement, the stroke may be increased by moving the reverse lever farther away from the center of the quadrant. In fact, the point of cut-off is generally controlled by local conditions, and it ranges from 6 to 20 inches.

The point of cut-off may be changed by altering the lap of the valve, and this may also be accomplished by changing the angular advance of the eccentric, and its throw, without changing the lap of the valve.

An early cut-off of the steam causes a restriction of the steam port opening, especially near the point of cut-off, where the current of the steam is rapid, and is followed by an early release and exhaust closure, which causes considerable increased compression.

An early cut-off is of material advantage, when an engine is operated at high-speed, because it allows more time for the exhaust steam to escape during the quick movement of the valve.

The increased compression resulting from an early cut-off may be considered an unavoidable evil and an inherent defect, or as a desirable feature and an incidental advantage, of the combination slide-valve; but the point of view depends entirely upon existing conditions and varying theories.

An increase of the travel of the valve would generally be considered the most effective remedy, or correction, for compression, but where an unbalanced valve is used, an extension of the valve will increase the unbalanced area and the friction proportionally. When a contraction, or limitation, of the valve travel is considered necessary, duplicate or supplementary ports may be used to advantage.

The contraction of the steam port opening is not of much importance unless the steam is cut-off at an early

point in the stroke of the piston, and the most objectionable feature is an early, or premature, release of steam.

The point at which cut-off occurs is shown by the small arrow, when the valve is traveling to the left, as indicated by the arrow A in Fig. 6.

Expansion.

Following the point of cut-off, the piston will be moved by only the expansive force of the steam. That is, the pressure of the steam confined in the cylinder behind the piston will force it forward, or backward, as the case may be. This will increase the space in which the steam is confined and there will be a fall in pressure approximately proportional to the increase in volume. Expansion will continue from the time cut-off begins until the valve opens for exhaust, that is, for only a small portion of the stroke; a portion equal to the total of the inside and the outside lap of the valve.

It must be apparent that if the opening of the valve for exhaust could be delayed so that expansion would continue until the pressure of the steam dropped to zero, practically all of the available energy of the steam would be utilized and converted into draw-bar pull. But such an arrangement would be impracticable for an ordinary cylinder, and in order to free the cylinders, the steam

must be released earlier, to allow the steam *time* to escape from the cylinder chamber before the piston begins its stroke in the opposite direction.

Compression.

The desire to eliminate compression, or to reduce it to a minimum, is prompted by the fact that compression shortens the effective power stroke of the engine. The advantage of eliminating this resistance is generally admitted, for, although a portion of the energy exerted in causing compression may be regained later on, it is evident that *only a part* of it can reasonably be expected; but even if all of the power were returned there would be a useless trading of energy for which no benefits would be obtained.

Of course compression assists the lead opening, because it partly fills the clearance space with steam pressure, helps to maintain the heat of the cylinder, prevents loss from steam condensation, and also obviates the shock which would otherwise be caused by the admission of high-pressure steam for the return stroke. It must be remembered, however, that no advantage is derived from pressure on the crank pins when the piston is within two inches of the end of its stroke, for it only increases friction and performs no useful work. In fact compression represents negative horse power developed by the closing of the exhaust cavity.

In high-speed engines compression is generally considered advantageous and essential, unless it produces a pressure equal to or higher than the initial pressure, but this occurs only a part of the time, usually when the engine is running at the higher speeds with cut-off at ¼ or less. At other times compression does not fill the clearance space and the shortage must be provided by live steam from the boiler. But it should be remembered that excessive compression, even in high-speed service, will cause an engine to ride hard, and, if there be any wear in the rods and boxes it will pound.

Compression commences at the point indicated by the small arrow *a* when the valve is traveling to the left as indicated by the large arrow A in Fig. 7.

Release, or Exhaust.

The point of release is just as important as the point of expansion and compression, and may be considered an aid of the two events, for each depends upon the other. If release occurs too early the benefits of expansion are sacrificed and compression will be delayed; on the other hand if release occurs too late the beneficial effect of expansion will be off-set by back pressure.

The valve should open for exhaust a little before the piston reaches the end of its stroke, to give the exhaust steam ample *time* to escape from the cylinder before the

piston begins its return stroke. If the exhaust steam was not discharged before the piston commenced its return stroke, the cylinders would become choked and each end would continually be working against the other end.

This event of the valve stroke, release, is reached when the exhaust edge of the valve reaches the exhaust edge of

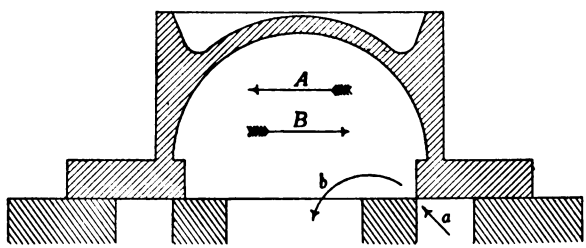


FIG. 7.

the steam port channel, which opens the port and permits the steam, which has been confined behind the piston to drive it through its working stroke, to escape to the atmosphere through the exhaust port.

The valve is in the same position for release as for compression, but it is moving in the opposite direction for the two events. The point of release is shown by the small arrow *b* when the valve is traveling to the right as indicated by the large arrow *B* in Fig. 7, and it is in this position when the engine is said to exhaust, or puff.

Travel.

It must be clear, even to the novice, that the travel of the valve is of necessity the most important event, or feature, of the valve operation, for all other events of the valve are controlled by the travel, and the slightest change in the travel of the valve must, therefore, result in a change in each of them.

The smallest amount of travel necessary to give a full port opening must equal twice the amount of the outside lap of the valve, plus twice the width of the steam port.

On the other hand, the total width of the lap of the steam port and the bridge combined, plus the over-travel, if any, should not be more than one-half of the travel.

In order to secure full steam port opening during any desired position of the stroke, the travel of the valve must exceed the amount of outside lap and the width of both steam ports.

The greater the travel the longer the steam port will remain open for steam admission. It should be remembered, however, that increased travel will require additional power to operate the valve, and will cause greater wear on the valve and its seat, but these disadvantages are partly overcome by the benefits resulting from better steam distribution.

Over-Travel.

The over-travel of a valve tends to choke the exhaust, but insures sufficient port opening with an early cut-off, and may be considered advantageous from the standpoint of steam distribution, but the gain obtained, or loss sustained, cannot be calculated by any fixed rule, for the results depend upon circumstances.

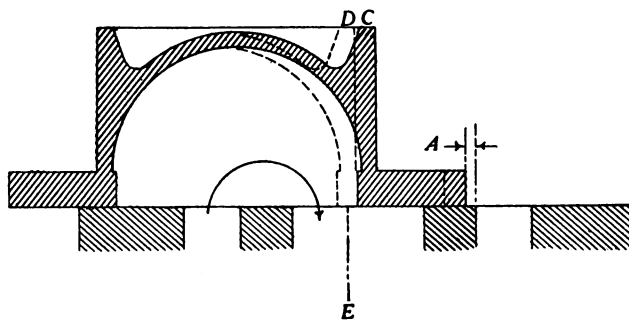


FIG. 8.

If over-travel causes the cut-off to occur too late, the evil effects can be overcome by increasing the outside lap, while delayed compression can be neutralized by increasing the inside lap; if exhaust occurs too late, as a result of over-travel, the inside lap may be sacrificed, but if there is no inside lap the valve may be given inside clearance.

The over-travel of a valve is shown by the space indicated by the letter A in Fig. 8.

It will be observed that the valve is shown in two positions on the valve seat. When in the position marked C the valve has traveled to the left beyond the point necessary to give the full steam port opening, and is about to commence its return travel to the right, but the inner edge of the valve has not traveled to the left beyond the center of the exhaust port. On the other hand, the position of the valve marked D shows that the inner edge has not only passed the center of the exhaust port, but the line marked E, which is drawn a distance from the left hand bridge equal to the width of the steam port. When a valve is given sufficient over-travel to force it beyond the position corresponding to the line E, or, in the position of the valve marked D, it is evident that the exhaust port will be contracted, and this will prevent the free escape of exhaust steam. Over-travel hastens admission, delays cut-off, release, and compression, lengthens admission, and shortens the expansion and compression periods.

Clearance.

It may be well for the reader to remember that there is a difference between *clearance,* and the terms *engine-clearance,* and *piston-clearance,* for they must not be confounded, but we shall here treat clearance as all of the space between the valve and the piston when the piston is at the beginning of its stroke.

The compression of steam by the advancing piston will be increased or decreased, and its expansive force is raised or lowered, in proportion to the amount of clearance present between the valve and the piston.

A reduction of the clearance space is a distinct advantage, aside from its connection with proper cushioning, for a reduction in this space adds to expansion, reduces the surface for steam condensation, and will prove economical, for the clearance space must be filled with live steam from the boiler to a considerable extent. While direct and straight ports to the cylinder will, to some extent, reduce the clearance, it is almost impossible to design, or construct, a large locomotive cylinder without allowing at least 8 or 9 per cent cylinder clearance.

Incidentally it may be said that *piston-clearance* is given to prevent the piston from striking and bursting either cylinder head, when the brasses on the connecting-rod wear and lost motion develops, or when water accumulates in the cylinder. In locomotive cylinders piston clearance varies from ¼ to ½-inch; it is generally ⅜ of an inch.

Back-Pressure.

As all of the steam employed in driving the piston cannot immediately escape through the port when the valve opens for exhaust, a certain portion of it remains in the cylinder for a time, and acts as an obstruction to

the piston when it begins its return stroke; the amount depending, to a considerable degree, upon the dimensions of the port and of the exhaust nozzle, together with the presence or absence of inside clearance of the valve.

There is always some back pressure in a locomotive cylinder, and a certain amount of it is necessary to create the draft, but an excessive amount will cause a serious loss of efficiency, and its presence can be readily detected by the increased amount of fuel consumed.

The retarding force of back pressure against the piston in locomotives with simple cylinders is about 7 to 9 pounds per square inch, but it may increase slightly as the speed increases, for the amount of condensation is not exactly constant at all speeds. It should not, however, be greater than 10 pounds at a speed of 20 miles per hour, or 15 pounds at 50 miles per hour.

Lead.

It is generally stated that lead is given to a valve in order that steam may be admitted *between* the piston and the cylinder head, just *before* the completion of the piston's stroke, to assist compression in cushioning, or gradually retarding, the piston, by tempering the sudden reversion and bringing the cranks smoothly over their centers, but such broad statements are incorrect and misleading.

As a matter of fact the advancing piston is brought to a state of rest by the steam compressed between the piston and the cylinder head, with the aid of preadmission; and the combined pressure of compression and preadmission in the piston clearance is sometimes equal to, if not greater than, initial pressure; as a result the steam admitted by lead can exert no force against the *piston* until the piston begins its return stroke, when the compressed steam expands.

Thus it is really preadmission, and not lead, which admits steam before the completion of the stroke. However, lead accomplishes this indirectly, for, without lead, there could be no preadmission.

Lead is given to increase the pressure in the clearance space of the steam port channel and to insure greater steam port opening for an abundance of full steam pressure, to assist expansion of the compressed steam behind the piston at the very beginning of the piston's stroke, when full steam pressure in the cylinder is most advantageous; it also helps to maintain a satisfactory supply of steam throughout the period of admission. Lead is not a part of the valve proper, but the point at which it begins is indicated by the letter A in Fig. 9.

No definite rule governing the proper amount of lead to be given in all cases can be stated, for the amount necessarily varies slightly with changes in speed, or the

amount of compression, and can only be accurately determined by the use of the Indicator; in fact, the amount of lead which would be proper and beneficial in one class of service might be detrimental in another kind of work.

For a long time it was the practice to allow from 3/16 inch negative to ⅛ inch positive lead, but of late years there has been a tendency to reduce the full gear lead, and the amount now employed varies from 3/12 to 1/16 of an inch.

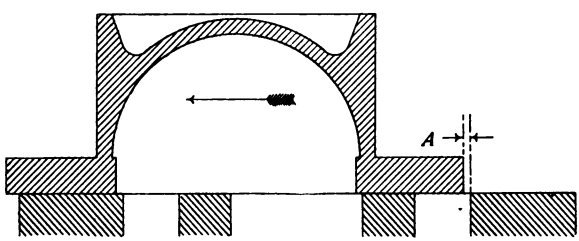


FIG. 9.

Good results can only be obtained by allowing just sufficient lead to fill the clearance space completely; an additional amount of lead would be wasteful of steam and prove unsatisfactory. For example, in starting an engine, or working it at slow speed, lead is undesirable, and should be sacrificed as much as possible, for it tends to retard the movement of the piston at the end of its stroke, and deprives the engine of a quick start. On the other hand, as the speed is increased, the reduced amount of time allowed for the admission of steam to the

cylinder renders a certain amount of lead necessary, and an increase of lead at high-speed is desirable and beneficial, because it imparts a larger port opening in the short cut-offs.

The beneficial results obtained by giving the valve lead have long been recognized, and we doubt whether any locomotive at the present time is operated without *some* lead.

It may also be noted that a proper reduction of the amount of lead will give a more efficient steam distribution, because such a reduction of lead will not necessarily reduce the maximum port opening at short cut-off sufficiently to materially impair the admission of steam by wiredrawing, for the speed of the piston at the quarter stroke is comparatively slow.

Lead is given to the valve by shifting the eccentric on the shaft to or from the crank-pins, the directing depending, of course, upon whether a direct or an indirect valve gear is used.

Lead increases when the reverse lever is moved toward the center, on an engine equipped with the Stephenson valve gear, because, as the lever is hooked up toward the center, both eccentrics influence the position of the valve. The lead of an engine equipped with the Stephenson valve gear can also be changed by moving the eccentrics toward or away from the crank-pins, depending

upon whether the lead is to be increased or decreased, or by reducing the lap of the valve if the eccentrics are to be retained in their normal position.

On the other hand the lead of an engine equipped with the Walschaert, or other radial type of valve gear, is constant, and can only be changed by changing the proportions of the combination, or lap and lead lever, or by reducing the lap of the valve. However, special "lead controllers," which will be described later on, are often employed to permit the control of lead with outside radial gears.

Even when the greatest care is exercised in designing and erecting a locomotive, a certain amount of lost motion in the driving boxes, and other working parts, soon develops, and the effect of such lost motion will result in delaying the movement of the valve if lead is not given, and, in some instances, will result in allowing the piston to begin its return stroke before steam is admitted to the cylinder. Of course this causes a certain amount of pounding, at each end of the piston stroke, if a proper amount of lead is not given.

Preadmission.

Preadmission is always subject to change, even when the lead is constant, and is, under ordinary conditions, the greatest factor in compression, for it commences when

the valve port opens for lead and ends when the crank pin is on center, or at the point where lead begins.

The accompanying outline drawing, Fig. 10, clearly shows the commencement and brief duration of preadmission.

By referring to the illustration it may be seen that preadmission commences in the right-hand port when the valve begins to uncover the port, when the crank-pin center, which is connected to the piston, is in the position shown; and it ends when the crank-pin is on dead center, that is, when the piston reaches the end of its stroke to the right.

This small movement of the piston is just sufficient to force the valve to the left to give full lead opening in the right-hand port.

Of course preadmission is the result of lead, but, during the time the main pin is traveling from the preadmission to the lead position, the movement of the valve is so rapid, while that of the piston is so slow, that preadmission may be considered negligible; at least, so far as starting is concerned. In fact, with certain valve gears, lead in full gear does not give more than 1/32 of an inch preadmission. But, on the other hand, when the reverse lever is moved toward the center of the quadrant, (hooked-up), from 7/8 to 1 1/4 of an inch preadmission

may be secured, but it may be as short as $\frac{5}{8}$ of an inch with other valve gears.

There is, however, considerable difference of opinion regarding the beneficial results to be obtained by preadmission, and many able men who have given the matter serious consideration are inclined to believe that preadmission is of no special benefit, for the reason that compression is fairly well developed before preadmission commences.

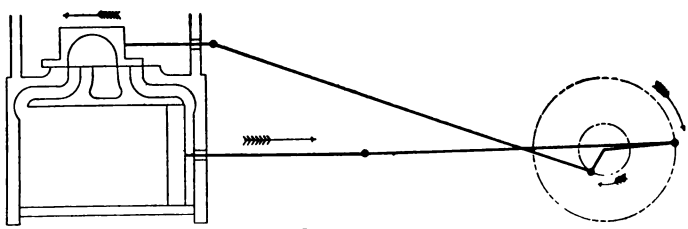


FIG. 10.

Wiredrawing.

As the amount of space available for the steam ports of a locomotive is limited, their area is necessarily small, even when they are fully opened, and their outline is far from being straight; as a result the passageways for the admission of steam from the boiler to the cylinder are restricted so that wiredrawing is unavoidable, to a certain extent. In such cases the steam is said to be throttled on its way from the boiler to the cylinder, and, in consequence, its pressure behind the piston is reduced

as the piston proceeds on its stroke, until, at the point of admission, it falls below the boiler pressure.

It is generally believed that wiredrawing in locomotive service is, to any extent, objectionable, and a thing to be avoided, but, as a matter of fact, it is always present, and renders useful service in certain classes of work. It aids in the maintenance of uniform boiler pressure for various rates of speed and is, therefore, beneficial from an economic standpoint.

For example, take an engine traveling up a grade at the rate of 20 miles per hour, with the throttle and reverse-lever in a position to require a moderate working load on the boiler; when the apex, or top, of the grade is passed, the load will be reduced in proportion to the grade, and the speed will increase twice or three times that maintained in ascending the grade, but the throttle and reverse-lever may be allowed to remain in their former positions, for the wiredrawing action will answer the demand, and prevent the cylinder from demanding more steam than the boiler can supply, without reducing the speed or maximum capacity of work.

In fact, wiredrawing results in economy of steam in proportion to the increase of speed from about 20 to 50 miles per hour, but for greater speeds the evil effect is increased, for it produces increased steam consumption.

Relative Positions of the Piston and Crank Pin.

As the connection between the crosshead and the piston is rigid, their positions correspond exactly; hence, if we know the motion of one of them, we also know the motion of the other. We will therefore dispense with a view of the piston in our illustrations for the present, and ignore the crosshead connection; proceeding on the theory that the piston is connected to the crank by a link. This link is generally termed a connecting rod, and is employed in changing the reciprocating rectilinear motion of the piston into the circular motion of the crank pin.

If the eccentric rod were of indefinite length, or if its obliquity were neglected, then, with the crank pin at any position between the front and back dead centers, the corresponding position of the piston could be ascertained by drawing a perpendicular line upon the diameter of the crank pin circle, which may be said to represent the stroke of the piston. For example, when the crank had traveled one-quarter of a revolution, the piston would be in the center of its stroke, etc. In practice, however, the changing angles of the connecting rod during the different periods of the crank pin revolution must be taken into consideration.

Any irregularities imparted by the crank pin to the motion of the piston will be conveyed into the motion of

the valve, but the throw of the eccentric rod is generally so small in comparison with that of the crank pin that it is inappreciable and may be disregarded, for the eccentric rod is proportionally longer than the connecting rod, and it follows that the distortions in the motion of the valve are necessarily much less than those in the motion of the piston.

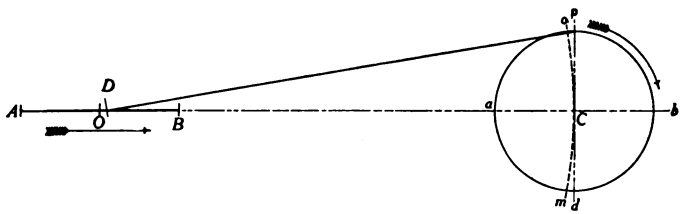


FIG. 11.

The obliquity, or angularity, of the connecting rod can be more forcibly impressed upon the reader's mind by the use of an illustration than by any written description we might give, so we shall proceed to construct Fig. 11, to show the travel of the piston, crank pin and connecting rod. The length of the connecting rod, which is measured from the center of the crank pin to the center of the crosshead pin, varies in practice from four to eight times the length of the crank, so, for the purpose of this illustration, we will use a connecting rod six times the length of the crank.

First we will draw the circle, *a p b d*, which is the crank pin circle, with the diameter equal to the travel

of the piston. Let *C* represent the center of the crank shaft, and the horizontal line, *A b*, which passes through the center of the circle *C*, the center line of motion of the crosshead, piston rod and piston. Now from the point *a*, of the circle as a center, and with a radius equal to the length of the connecting rod, strike an arc cutting the line *A b* in the point *A*. This point will be the center of the crosshead pin at the forward end of its travel. From the point *b* as a center, and with the same radius, strike another arc cutting the line *A b* at the point *B*, and this point will be the center of the crosshead pin at the back end of its travel. Since the distance between the points *A* and *B* represent the full length of the piston stroke, we may find the center of the stroke by striking an arc on the line *A b* from the center *C* of the circle, which will equally divide the distance between *A* and *B;* and we will mark the point *O* on the line *A b*. Now from the point *O*, and with a radius equal to the length of the connecting rod, we will draw the arc *o C m*, and mark the points *o* and *m* where it intersects the crank pin circle; the two points *o* and *m* will be the centers of the crank pin when the piston is at the center of its stroke.

By referring to the illustration again it will be noticed that the end of our connecting rod has reached its quarter stroke and is at the point *p*, so, with a radius

equal to the length of the connecting rod, we will strike an arc cutting the line *A* b in the point *D*. The point *p* bisects the semi-circumference which extends from *a* to *b*, but it may be readily observed that the point *D* does not bisect the stroke *A B;* in fact the point *D* is much nearer the *B* end of the stroke than it is to *A*.

From this we see that during the first half stroke of the piston the crank pin travels from *a* to *o*, and during the second half stroke of the piston the crank pin travels from *o* to *b*. Since the motion of the crank must be uniform, the average speed of the piston during the first half stroke will be a little greater than during the second half stroke; and this difference in the speed of the piston is due to the obliquity of the connecting rod.

Now, assuming that our illustration of the variation in the piston's travel, between the first and the second half of its stroke, is clear, the reader may find it profitable to ascertain the movements in the different periods of the stroke.

To assist in the continuation of our examination, let us make a new illustration, making the crank pin circle and piston travel the same as in Fig. 11, but on a slightly larger scale. We shall, however, employ, in Fig. 12, a connecting rod three times the length of the crank, instead of six times its length, as used in Fig. 11. A connecting rod of this length could not be used in loco-

motive service, and it is only adopted here to emphasize the movement of the piston at different positions.

Our next step will be to divide the piston travel and the upper semi-circumference of the crank pin circle into eight equal parts, which we will number to correspond. Advancing the short connecting rod from the point *a*, which represents the front dead center, to the

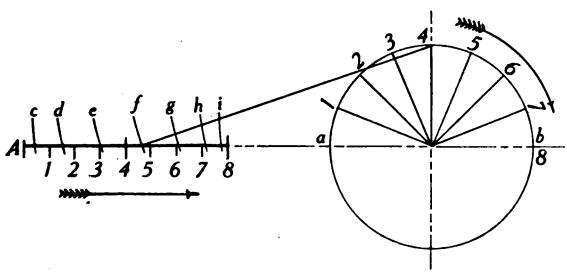


FIG. 12.

point 1 on the semi-circumference of the crank pin circle, we strike an arc cutting the line *A b* in the point *c*, which will indicate the position of the crosshead pin, and, in a similar manner, from each of the other figures on the semi-circumference of the crank pin circle we strike similar arcs and mark with letters to correspond.

The crank pin moves at a symmetrical rate of speed through the divisions we have marked on the semi-circumference of the crank pin circle, yet we find from an examination of Fig. 12, that the movements of the crosshead pin, and therefore of the piston, from *c* to *d*,

and from *e* to *f*, on the line *A b*, are not equal. In fact, it shows that the piston will, at the commencement of its stroke, move comparatively slow, and increase its speed as it approaches the center of the stroke, and when the piston is moving away from the center of the stroke, its speed is constantly decreasing.

This variable motion of the piston is caused chiefly by changing the rectilinear, or straight, motion into a uniform circular motion, and to some extent, by the angle formed by the center line of the connecting line and the horizontal line *A b*, an angle which is constantly changing during the stroke.

Various expedients have been adopted to overcome, or neutralize, the evil effects due to the angularity of the connecting rod, but it has been found that fairly good results can be obtained, at small expense, by setting the link saddle pin back.

Relation Between the Valve and Eccentric, or Crank Arm.

Thus far we have confined our discussion to the valve proper, and the valve events, referring to the connection of the valve with the cylinder and piston to a limited extent, and with but casual reference to the parts by which the movement of the valve is controlled.

We now feel that the reader should begin the study of the eccentric, or crank, and the relative position of

it to the corresponding movements of the valve. Following our original plan we shall first look at the most early and most common methods of transmitting the reciprocating motion of the piston to the valve through the rotating motion of the eccentric.

The Eccentric.

Literally, the word eccentric means, "out from the center." While there is not much similarity in the appearance of an eccentric and a crank arm, they are used for the same purpose, and give exactly the same results. It may also be said that there is, in reality, nothing mysterious about an eccentric, for it is essentially a crank, whose length is equal to the radius of the throw of the eccentric with the pin increased to include the shaft.

A crank arm can only be used at the end of a shaft, while an eccentric may be attached at any point desired along the shaft, without reducing (by cutting or dividing) the strength of the shaft, and it is the latter feature which often decides in a selection between the two.

To make the subject more clear we present two views of the eccentric, and one of a common crank, on page 84. It will be observed that Fig. 14 shows the eccentric in full, while Fig. 13 shows it in section, and Fig. 15 shows a common crank.

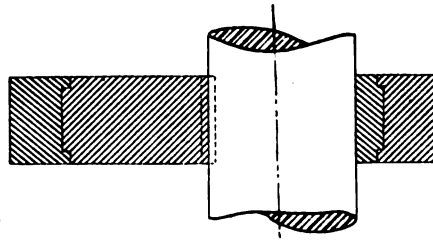


FIG. 13.

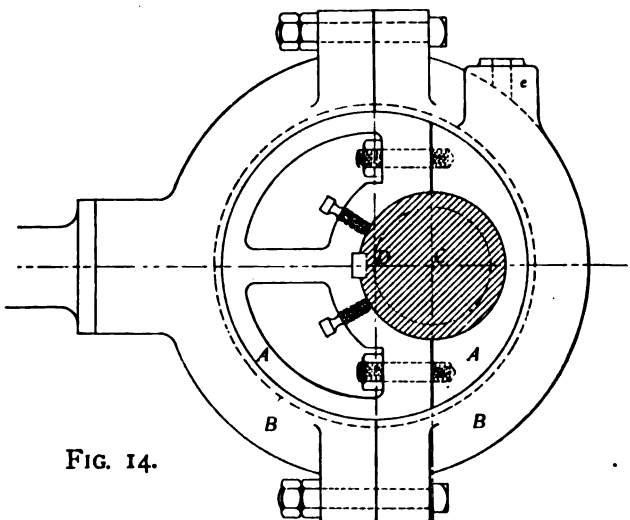


FIG. 14.

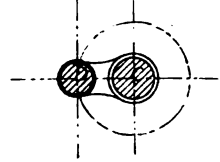


FIG. 15.

In Fig. 14, the eccentric proper, *A A,* which rotates with the shaft and corresponds to the crank arm, is called the eccentric sheave. The eccentric sheave is surrounded by a metal strap, or band, commonly called the eccentric strap, *B B,* inside of which the eccentric sheave is free to turn. Both the sheave and the strap are made in two halves, so that they can be readily put on or taken off; the two pieces of the sheave are parted on a line passing through the center of the shaft at right angles to the horizontal center line of the eccentric, and are bolted together and rigidly keyed or fastened by studs to the shaft. The strap is grooved so as to prevent it from becoming displaced or getting off the sheave, and it is bolted together at its top and bottom.

An oil cup, *e,* Fig. 14, is usually cast solid on one half of the strap, for particular care must be given to the lubrication, the friction of eccentrics being much greater than that of cranks because they have larger sliding surfaces, between the sheave and the strap. It may also be noted that an eccentric requires more metal for its construction and space for its operation than a crank; it also absorbs more power than the crank, as a result of the greater leverage at which the friction acts, and it is used in preference only where the throw is comparatively short.

A crank is used for converting the rectilinear motion of the piston into rotary or circular motion, while an eccentric is usually employed for converting the rotary or circular motion of the shaft back into the straight motion of the valve.

When the valve receives its motion from an eccentric, the periods of steam distribution are necessarily controlled by the position of the eccentric, and they occur earlier or later in the stroke as the eccentric is turned forward or backward on the shaft.

Now, referring to Fig. 14, we find that the point C is the center of the shaft, and the point D is the center of the eccentric sheave. As the sheave revolves with the shaft the point D will describe the small dotted circle on the shaft, whose center is the center of the shaft. As a result when half of a revolution is completed, the eccentric strap, *B B*, will be moved horizontally a distance equal to the diameter of the dotted circle, which equals the *throw* of the eccentric, or the travel of the valve. The distance between the center of the shaft C and the center of the sheave D equals the length of the crank arm, and is termed the *radius* or *eccentricity* of the eccentric, which is one-half of the *throw*. All movements of the eccentric are transmitted to the valve by means of an eccentric rod and valve stem, which connect the two.

In practice, the travel of a locomotive valve does not always equal the throw of the eccentric; the difference may be attributed to the influence of the link, and, in many cases, to the unequal length of the rocker arms.

With this brief description of the eccentric, and explanation of its use, we shall proceed to develop its relation to, and connection with, the movement of the valve.

Position of the Eccentric for a Valve Without Lap.

In practice the eccentric is always mounted on the crank-shaft, and, if the valve has no lap and no lead, the center line of the eccentric must be perpendicular to the center line of motion of the valve gear when the piston is at end of its stroke and the valve stands in its central position.

The eccentric must, in all cases, travel ahead of the crank, and when the connection between the valve and the eccentric is direct, as in Fig. 16, and the piston is at the end of its stroke, the crank is on dead center; therefore, the valve must be in its central position, and the center of the eccentric must be midway between the two dead centers.

In such a case, when the piston begins to move toward the right-hand end of the cylinder, the center of the

eccentric will turn, with the crank, to the right, and pull the valve in the same direction, so that the left-hand edge of the valve face will open the left-hand steam port. This admits steam to the left-hand steam port, behind the piston, and drives it in the desired direction, while the right-hand edge of the valve will uncover the right-hand steam port, and permit the steam in front of the advancing piston to escape.

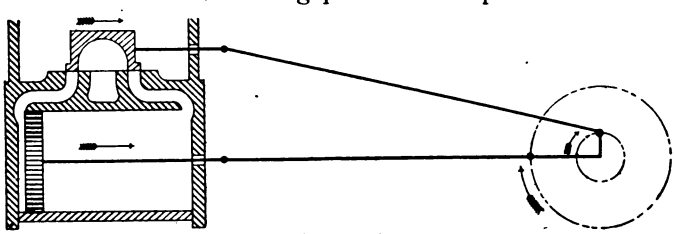


FIG. 16.

If the center of the eccentric was placed one-quarter behind the crank, midway between the two dead centers, it is apparent that when the piston commenced its stroke to the right the valve would be moved to the left, thereby uncovering the right-hand steam port for the admission of steam in *front* of the piston. If, however, we turn the crank in the opposite direction, that is, to the left, the valve will uncover the left-hand steam port and admit steam *behind* the piston and the engine will move in the opposite direction. In other words, if the center of the eccentric is placed 90 degrees *ahead*

of the center line of motion of the valve gear the shaft will rotate to the *right;* if placed 90 degrees *behind* the center line of motion of the valve gear the shaft will turn to the *left.*

We shall assume that the parts, shown in Fig. 16, are about to move in the direction of the arrows. When the crank reaches the top quarter, the eccentric center will be on the back dead center, the valve will have reached the end of its travel to the right, and the piston will have almost reached the center of its stroke. The piston would be exactly at the center of its stroke but for the angularity of the connecting-rod, which has been explained. When the parts have moved another quarter the crank will be on the back dead center, the eccentric will be on the lower quarter, the piston will have reached the end of its stroke to the right, and the valve will have moved to the left, so that it stands central. Another quarter movement will place the crank on the lower quarter, the eccentric center on the forward dead center, the valve at the end of its travel to the left, and the piston almost in the center of its stroke. The final quarter movement will place the parts in the positions they originally occupied in the illustration. (It must be obvious that a valve of this pattern will not admit steam into the cylinder to *start* the movement of

the engine from a state of rest, for the valve opens just *after* the stroke of the piston commences.)

The primitive form of valve illustrated, which just covers the steam ports when in mid-position, can not be used with economy, as it allows steam to follow the piston for the whole of the stroke, and does not permit the use of its expansive properties, for the simple reason that, at the instant the admission of steam ceases, the exhaust of the same body of steam must immediately commence, thereby sacrificing the beneficial results obtained from compression.

As it has been shown that this elementary valve is extremely wasteful, and therefore impractical, let us modify it by extending its face so that when the valve is in mid-position it will overlap the steam ports. Thus we may see what effect the addition of lap to the valve will have upon the position of the eccentric.

Position of the Eccentric for a Valve With Lap.

Now that we are about to examine the positions of the eccentric for a valve with lap, it may be well to again refer to our definitions of the terms "linear advance" of the valve, and "angle of advance" of the eccentric. It will be found that the positions of the two are closely related, and, in a sense, dependent upon each other, yet

the distinction between the terms is of great importance and should be thoroughly understood.

When the valve is given lap, that is, when the face of the valve is lengthened, it will be necessary for the linear

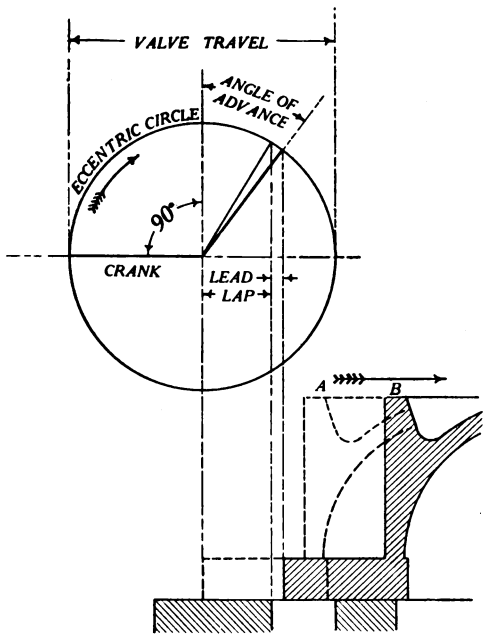


FIG. 17.

travel of the valve to be increased a distance equal to the outside lap, for the admission of steam to take place. The valve must travel the length of the lap plus the width of the port, if the port is to be fully opened, and, of

course, increased linear travel means increased throw, or advance of the eccentric, which will require a change in the position of the eccentric on the crank shaft.

In Fig. 17 we have shown the valve in two positions, and placed the eccentric circle above the same, so that the reader may readily observe the effect that a movement of either the valve or the eccentric will have upon the other. The dotted lines represent the valve standing in the center of its travel, and, for reference, we have marked it *A*. The other view shows the valve at the commencement of the piston stroke, and is marked *B*.

If we assume that the piston is at the beginning of its stroke to the right, it is obvious that the valve *A* is not in a position to admit steam to the left-hand port; in fact to admit steam into the cylinder behind the piston the valve *A* must be moved to the right from its central position until it has opened the left-hand port, and, if the valve is given lead, an additional amount equal to the lead. As a result the center of the eccentric, at the beginning of the piston's stroke, cannot remain in the position shown in Fig. 16, that is, at right angles to the crank, but must be advanced (in this case, to the right) an additional amount, equal to the lap and the lead combined, as shown in the illustration. When the necessary amount of linear advance is given to the valve *A*, it will occupy the position of the valve *B* and admit steam

behind the piston at the beginning of its stroke to the right.

As a result of the change in the position of the eccentric all of the events effected by the valve will be completed earlier; that is, the port for the admission of steam will be partly opened at the beginning of the piston's stroke, and it will be closed before the end of the stroke, so that the steam will be allowed to expand during the period from cut-off to release. The exhaust will also be closed before the end of the stroke, and the undischarged steam trapped in the cylinder will undergo compression.

Relative Positions of the Valve and Piston.

Now that the reader is familiar with the various positions of the valve on its seat for the events of the stroke, and the position of the eccentric and crank for a valve without lead or lap, let us compare the different positions of a valve with lap to the corresponding positions of the piston.

The motion of the piston is not symmetrical, as has already been explained, but we shall here disregard this slight difference due to the angularity of the connecting-rod.

Admission, or lead opening. In Fig. 18 the slide valve is shown at the point of lead opening, that is, the left-hand port is open to the admission of steam into the head-

end of the cylinder. These positions are shown on a larger scale in Figs. 6 for admission, and 9 for lead. The eccentric arm, instead of being at right angles to the crank at the beginning of the stroke, as shown in Fig. 16, is set ahead an amount equal to the angular advance. The reason for this is, as previously explained, that the valve, instead of standing central as it would if it had no lap or lead, must be advanced, or set ahead

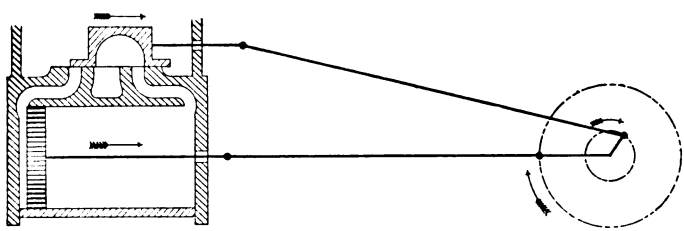


FIG. 18.

an amount equal to the outside lap plus the lead, in order that steam may be admitted into the cylinder at, or a little before, the commencement of the stroke of the piston. The crank pin is on the forward dead center, and the piston is about to begin its stroke, so the admission of steam behind the piston will now begin, and continue, through the left-hand port, behind the piston, until the valve returns to close the left-hand port.

Cut-off. When the valve has traveled to the end of its stroke to the right, in order to fully open the left-

hand port to admission, it must return in the direction of the arrow to the position shown in Fig. 19. The eccentric arm will have moved from its position in Fig. 18 to that in Fig 19, in the direction indicated by the arrow. During this period a corresponding movement of the crank takes place, causing the piston to assume the position shown in Fig. 19. It will be noticed that the piston has already completed more than half of its

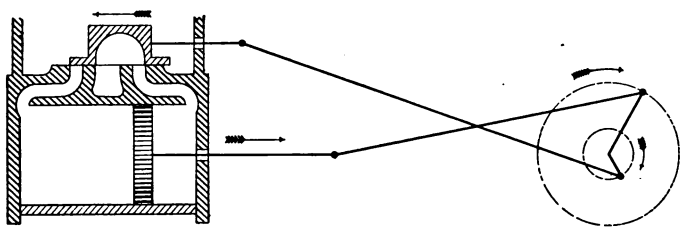


FIG. 19.

stroke to the right. A further movement of the valve in the direction of the arrow will close the left-hand port to the admission of steam, that is, cut-off takes place. This point is more clearly illustrated in Fig. 6. Expansion will now commence in the head-end of the cylinder, and the force of the expanding steam will cause the piston to continue its movement to the right.

Compression. As the position of the piston in Fig. 20 is reached, the valve will have moved to cover the right-hand port, as shown in the illustration, and also in Fig. 7. The unexhausted steam trapped in the crank

end of the cylinder will be compressed by the expansive force of the steam previously admitted to the head-end of the cylinder acting upon the piston and moving it to the right. Expansion, theoretically, should cease immediately at the point of release, but in practice there is a gradual, although rapid, decrease in pressure on the piston, causing a slight prolongation of expansion.

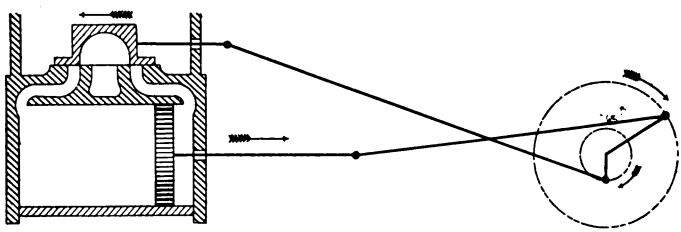


FIG. 20.

Release. Because the valve shown in these illustrations is line-and-line with the steam ports, that is, it has no inside lap or clearance, the point of the commencement of compression in the crank-end corresponds to the point of release in the head-end. If the valve had inside lap, release would follow compression; if it had clearance, release would take place before compression began. This is readily understood by referring to Fig. 20. Release is also shown in Fig. 7 on an enlarged scale. Exhaust will continue from the point of release shown in Fig. 20 at the left-hand port, until the point of compression on the return stroke of the piston.

Admission,—Crank-end. Compression will continue from the point shown in Fig. 20 to that shown in Fig. 21, where the valve is just about to uncover the right-hand port to admission. It may be seen from the position of the crank that the piston has not yet reached the end of its stroke. The reason for this is that pre-admission commences a little before the point of full lead opening, and therefore before the completion of the stroke, is

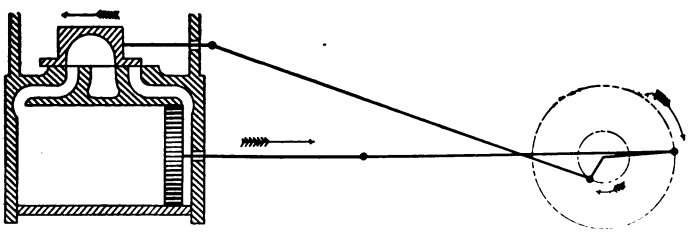


FIG. 21.

reached. The eccentric arm, too, must travel an amount equal to the lead opening of the valve in order to reach a point exactly opposite that shown in Fig. 18.

Valve With Rocker Arrangement.

The reader should bear in mind that the foregoing illustrations are primarily intended to show the corresponding positions of the piston and a valve with lap, and the various positions of the eccentric and crank are shown only for that purpose.

It may be observed in the illustrations that the valve is not in line with the eccentric, and for this reason, if

for no other, the direct connection between the valve and eccentric would be impracticable. As a result of this angle between the eccentric rod and the center line of motion, the speed of the valve would vary during the stroke, even though the eccentric rotated at a constant speed. For this reason a rocker, of the type shown in Figs. 22 or 23, is interposed between the eccentric and the valve, and is used merely to connect the eccentric rod and the valve

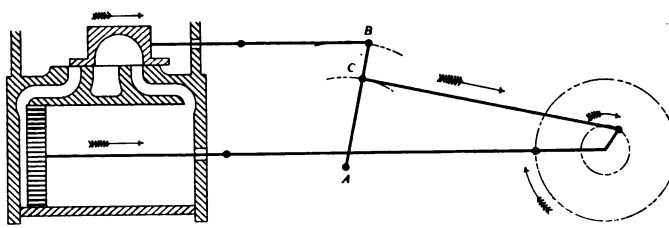


FIG. 22.

rod. While it affects the position of the eccentrics, and often the travel of the valve, the rocker will not necessarily affect the rules regarding the valve construction.

In Fig 22 the rocker is pivoted at A, and the rotating motion of the eccentric imparts a backward and forward motion to the rocker through the connection C. This reciprocating movement is applied to the valve through the valve stem connected to the rocker at B.

It is often more convenient to place the pivot of the rocker between the valve stem connection and the connection with the eccentric rod, as shown in Fig 23 at

the point A. Now, in order to impart to the valve the same motion as that in Fig 22, the eccentric must be set ahead 180 degrees, for it will be seen, as indicated by the arrows, that in this illustration the valve stem and eccentric rod move in opposite directions. Thus, even though the eccentric rod moves in the direction opposite to that in Fig. 22, the valve rod moves in the same direction as formerly, because the pivot A of the

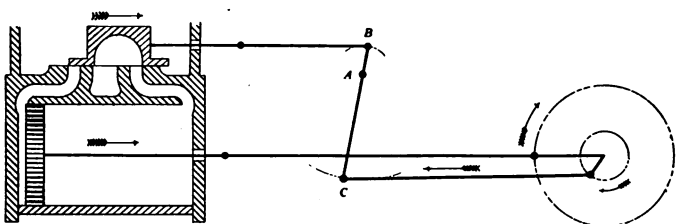


FIG. 23.

rocker is placed *between* the two connections. Either type of rocker may therefore be used with the same results.

The rocker may also be used so as to make the throw of the eccentric less than the travel of the valve, for, as the distance between the points B and C, Fig. 22, is increased, the throw of the eccentric for a given travel of the valve is decreased.

Direct and Indirect Valve Motion.

The rocker arm is usually the means by which the motion imparted to the eccentric is reversed. With the

Stephenson link motion, if both arms extend in the same direction from the rocker shaft, the motion is direct. However, if they extend in opposite directions from the shaft, one up and one down, the motion is reversed, and the valve is indirect.

With the more modern radial valve gears, the motion is direct when the link block is working below the center of the link, when it has the same direction of motion from the eccentric to the valve. But when the link block is working above the center of the link, the motion is indirect. Therefore, reversing the engine, or placing the link block above the center of the link, changes the valve motion from direct to indirect.

Inside admission valves, as the modern piston valve, also influence the valve motion. When the valve admits steam from the inside, it must move in the opposite direction to the outside admission valve, that is, in a direction opposite to that of the piston at the commmencement of the piston stroke. To accomplish this, the valve may be driven direct, without a rocker, leaving everything else the same. The rocker will give the desired motion, however, if both eccentrics are moved half way round the axle.

The following is a simple means of distinguishing between direct and indirect motion:

If the crank pin and eccentric are on the same side of the driving shaft, and a rocker is provided to reverse the motion, the valve is indirect, outside admission. But if the crank pin and eccentric are together without a rocker arm, the valve has inside admission, direct.

If the crank pin and eccentric are on opposite sides of the drifting shaft, with a rocker to reverse the motion, the valve has inside admission, and is indirect. With crank pin and eccentric opposite without a rocker, however, the valve has direct motion, and is outside admission.

The distinction may be stated briefly as follows: When the eccentric rod moves forward, and produces a forward movement of the valve, the motion is direct. But, if when the eccentric moves forward, the valve moves back, the motion is indirect.

Problems Relating to Lap of the Slide Valve.*

To find the point of cut-off when the lap and travel of the valve are given, the valve to have no lead.

Example.—Lap of valve is one inch; travel, 5 inches; no lead; stroke of piston, 24 inches. At what part of the stroke will the steam be cut off

We must first find the center *c*, Fig. 24, of the circle *a b m*, whose circumference represents the path of the

*This article by the late J. G. A. Meyer, M. E., in The Locomotive Up-to-Date, is reproduced by permission of the publishers.

center of eccentric, and this is found by placing the valve in a central position, as shown in dotted lines in this figure. Then the edge *c* of the valve will be the center of the circle. The valve drawn in full lines shows its position at the commencement of the stroke of piston; and since the valve is to have no lead, the edge C_2 will

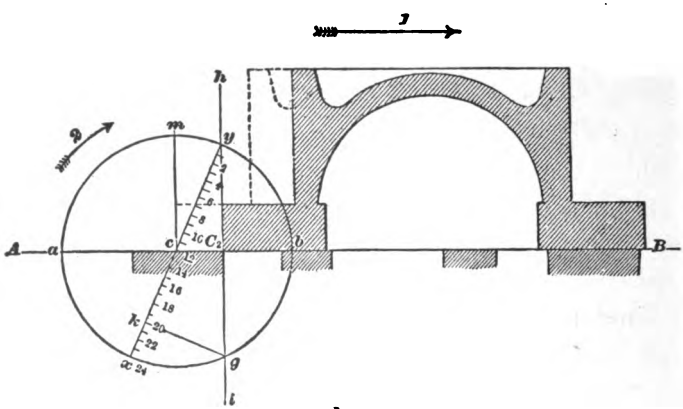


FIG. 24.

coincide with the outer edge of the steam port. Through the edge C_2 draw the line *i h* perpendicular to the line *A B;* the line *i h* will intersect the circumference *a b m* in the point *y,* and this point will be the center of eccentric when the piston is at the beginning of its stroke. Now, assume that the circumference *a b m* also represents, on a small scale, the path of the center of the crank-pin; then the diameter *y x* of this circle will repre-

sent the length of the stroke of the piston; the position of this diameter is found by drawing a straight line through the point *y* (the center of the eccentric when the piston is at one end of its stroke) and the center *c*. Also assume that the point *y* represents the center of the crank-pin when the piston is at the beginning of its stroke. To make the construction as plain as possible, divide the diameter *y x* into 24 equal parts, each representing one inch of the stroke of piston, and for convenience number the divisions as shown. The arrow marked 1 shows the direction in which the valve must travel, and arrow 2 indicates the direction in which the center *y* must travel. Now it must be evident, because the points *y* and C_2 will always be in the same line, that during the time the center *y* of the eccentric travels through the arc *y g*, the valve not only opens the steam port, but, as the circumference *a b m* indicates, travels a little beyond the port, and then closes the same, or, in short, during the time the center of eccentric travels from *y* to *g*, the port has been fully opened and closed; and the moment that the center of eccentric reaches the point *g*, the admission of steam into the cylinder is stopped. We have assumed that the point *y* also represents the position of the center of crank-pin at the beginning of the stroke; and, since the crank and eccentric are fastened to the same shaft, it follows that during the time

the center of eccentric travels from y to g the crank-pin will move through the same arc, and when the steam is cut off the crank-pin will be at the point g. Therefore, through the point g draw a straight line $g\,k$ perpendicular to the line $y\,x$; the line $g\,k$ will intersect the line $y\,x$ in the point k, and this point coincides with the point marked 20; hence the steam will be cut off when

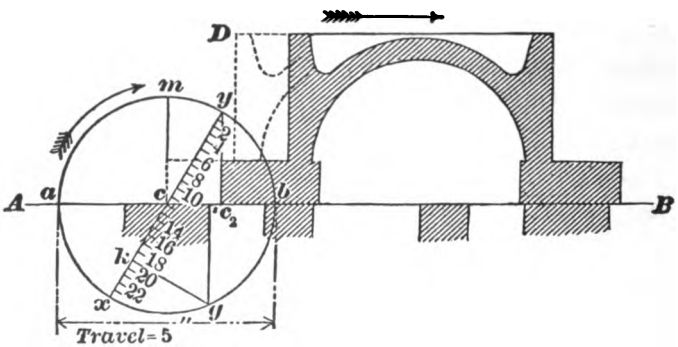


FIG. 25.

the piston has traveled 20 inches from the beginning of its stroke.

Lead Will Affect the Point of Cut-Off.

In Fig. 24 the valve had no lead; if, now, in that figure, we change the angular advance $m\,y\,c$ of the eccentric so that the valve will have lead, as shown in Fig. 25, then the point of cut-off will also be changed. How to find the point of cut-off when the valve has lead, is shown in Fig. 25.

Example.—The lap of valve is 1 inch, its travel 5 inches; lead ¼ of an inch (this large amount of lead has been chosen for the sake of clearness in the figure); stroke of piston, 24 inches; at what part of the stroke will the steam be cut off?

On the line *A B*, Fig. 25, lay off the exhaust and steam ports; also on this line find the center *c* of the circle *a b m*, in a manner similar to that followed in the last construction, namely, by placing the valve in a central position, as shown by the dotted lines, and marked *D*, and then adopting the edge *c* of the valve as the center of the circle *a b m*; or, to use fewer words, we may say from the outside of the edge *s* of the steam port, lay off on the line *A B* a point *c* whose distance from the edge *s* will be equal to the lap, that is, 1 inch. From *c* as a center, and with a radius of 2½ inches (equal to one-half of the travel), describe the circle *a b m*, whose circumference will represent the path of the center of eccentric. The lead of a valve in a locomotive is generally 1/32 and sometimes as much as 1/16 of an inch, when the valve is in full gear, but for the sake of distinctness we have adopted in this construction a lead of ¼ of an inch in full gear. Draw the section of the valve, as shown in full lines, in a position that it will occupy when the piston is at the beginning of its stroke, and consequently the distance between the edge c_2 of the valve and the edge *s*

of the steam port will, in this case, be ¼ inch. Through c_2 draw a straight line perpendicular to *A B*, intersecting the circumference *a b m* in the point *y;* this point will be the center of the eccentric when the piston is at the beginning of its stroke, and since it is assumed that the circumference *a b m* also represents the path of the center of the crank-pin, the point *y* will also be the position of the crank-pin when the piston is at the commencement of its stroke. Through the points *y* and *c* draw a straight line *y x*, to represent the stroke of the piston, and divide it into 24 equal parts. Through the point *s* draw a straight line perpendicular to *A B*, intersecting the circumference *a b m* in the point *g*, through *g* draw a straight line perpendicular to *y x*, and intersecting the latter in the point *k;* this point will be the point of cut-off. If, now, the distance between the point *k* and 19 is about ⅛ of the space from 19 to 20, we conclude that the piston has traveled 19⅛ inches from the beginning of its stroke when the admission of steam into the cylinder is suppressed.

Here we see that when a valve has no lead, as in Fig. 24, the admission of steam into the cylinder will cease when the piston has traveled 20 inches; and when the angular advance of the eccentric is changed, as in Fig. 25, so that the valve had ¼ of an inch lead, the point of cut-off will be 19⅛ of an inch from the beginning of the

stroke, a difference of ⅞ of an inch between the point of cut-off in Fig. 24 and that in Fig. 25. But the lead in locomotive valves in full gear is only about 1/32 of an inch, which will affect the point of cut-off so very little that we need not notice its effect upon the period of admission, and, therefore, lead will not be taken into consideration in the following examples.

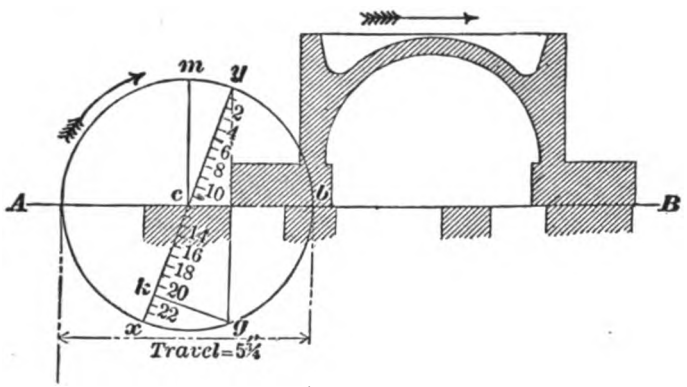


FIG. 26.

The Travel of the Valve Will Affect the Point of Cut-Off.

Fig. 26 represents the same valve and ports shown in Fig. 24, but the travel of the valve in Fig. 26 has been increased to 5¾ inches. The point of cut-off *k* has been obtained by the same method as that employed in Figs. 24 and 25, and we find that this point *k* coincides with point 21. Now notice the change caused by an increase

of travel; when the travel of the valve is 5 inches, as shown in Fig. 24, the admission of steam into the cylinder will cease when the piston has traveled 20 inches from the commencement of its stroke, and when the travel of the same valve is increased ¾ of an inch, as shown in Fig. 26, the admission of the steam will not be suppressed until the piston has traveled 21 inches. Here we notice a difference of 1 inch between the two points of cut-off. But it must be remembered that when the travel of a valve for a new engine is to be found or established, the point of cut-off does not enter the question; we simply assign such a travel to the valve that steam ports will be fully opened, or give it slightly greater travel when the valve is in full gear. The point of cut-off is regulated by the lap and position of the eccentric.

In order to find the point of cut-off it is not necessary to make a drawing of the valve, as has been done in Fig. 25. The only reason for doing so was to present the method of finding the point of cut-off to the beginner in as plain a manner as possible. In order to show how such problems can be solved without the section of the valve, and, consequently, with less labor, another example is introduced.

Example.—Lap of valve is 1⅜ inches; travel 5½ inches; stroke of piston, 24 inches; width of steam port, 1¼ inch; find the point of cut-off.

Draw any straight line, as *A B*, Fig. 27, anywhere on this line mark off 1¼ inch, equal to the width of the steam port. From the edge *s* of the steam port lay off in the line *A B* a point *c*, the distance between the points *s* and *c* being 1⅜ inches; that is, equal to the amount of lap. From *c* as a center, and with a radius equal to half the travel, namely, 2¾ inches, draw a circle *a b m*; the circumference of this circle will represent the path of the eccentric, and also that of the crank-pin. Through *s* draw a straight line *i h* perpendicular to *A B*; this line *i h* will intersect the circumference *a b m* in the points *y* and *g*. Through the points *y* and *c* draw a straight line *y x*; the diameter *y x* will represent the stroke of the piston. Divide *y x* into 24 equal parts; through the point *g* draw a straight line *g k* perpendicular to *y x*, and intersecting *y x* in the point *k*, and this point is the point of cut-off. Since *k* coincides with the point 18, it follows that the piston had traveled 18 inches from the beginning of its stroke when the flow of the steam into the cylinder ceased.

Now we may reverse the order of this construction and thus find the amount of lap required to cut off steam at a given portion of the stroke.

Example.—Travel of valve is 5¾ inches; stroke of piston, 30 inches; steam to be cut off when the piston

has traveled 22 inches from the beginning of the stroke; find the lap.

Draw a circle $a\,b\,m$ whose diameter is equal to the travel of the valve, viz., 5¾. Through the center c draw the diameter $y\,x$. In Fig. 28 we have drawn the line $y\,x$ vertically, which was done for the sake of convenience; any other position for this line will answer

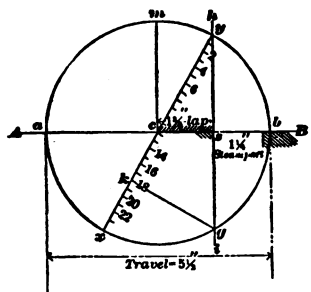


FIG. 27.

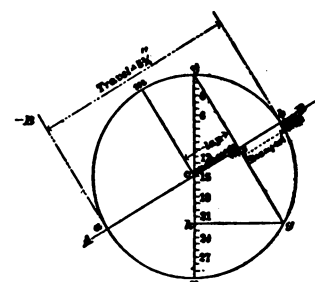


FIG. 28.

the purpose equally well. The circumference $a\,b\,m$ represents the path of the center of the eccentric, also that of the crank pin; the diameter $y\,x$ will represent the stroke of the piston, and, therefore, is divided into 30 equal parts. The steam is to be cut off when the piston has traveled 22 from the beginning of the stroke, therefore, through the point 22 draw a straight line $g\,k$ perpendicular to $y\,x$, the line $g\,k$ intersecting the circumference $a\,b\,m$ in the point g. Join the points y and g by a straight line. Find the center s of the line $y\,g$, then,

through *s* and perpendicular to the line *y g*, draw the line *A B;* if the latter line is drawn accurately it will always pass through the center *c*. The distance between the points *s* and *c* will be the amount of lap required, and in this example it is 1-7/16 inch.

It sometimes occurs, in designing a new locomotive, and often in designing stationary or marine engines, that only the width of steam port and point of cut-off is known, and the lap and travel of the valve is not known. In such cases both of these can be at once determined by the following method:

Example.—The width of the steam port is 2 inches; the stroke of piston, 30 inches; steam to be cut off when the piston has traveled 24 inches from the beginning of its stroke; find the lap and travel of the valve.

Draw any circle, as *A B M*, Fig. 29, whose diameter is larger than the travel of the valve is expected to be. Through the center *c* draw the diameter *y x*, and, since the stroke of piston is 30 inches divide *y x* into 30 equal parts. Steam is to be cut off when the piston has traveled 24 inches; therefore through point 24 draw a straight line *g k* perpendicular to the diameter *y x*, intersecting the circumference *A B M* in the point *g*. Join the points *y* and *g* by a straight line; through the center *s* of the line *y g* draw a line *A B* perpendicular to *y g*. So far, this construction is precisely similar to that shown

in Fig. 28, and in order to distinguish this part of the construction from that which is to follow, we have used

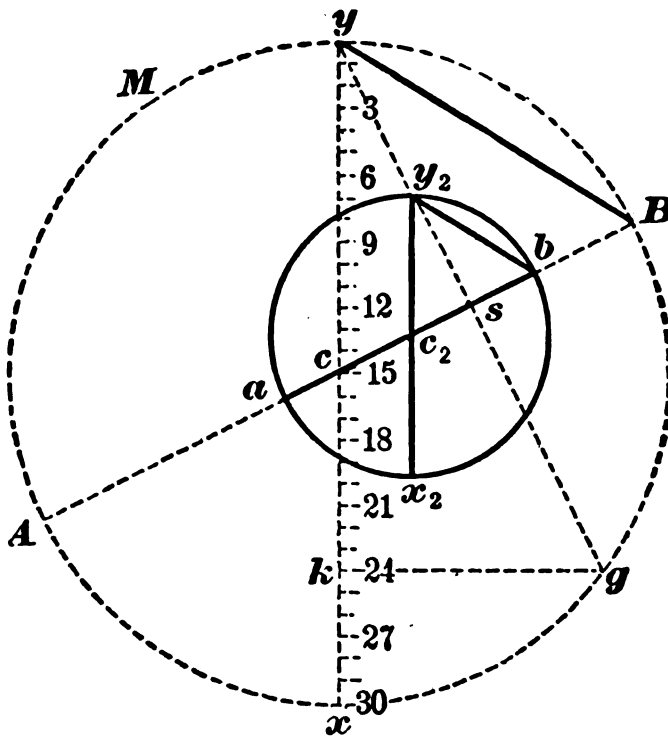


FIG. 29.

dotted lines; for the rest full lines will be used. It will also be noticed by comparing Fig. 29 with Fig. 28 that, if the diameter *A B* had been the correct travel of valve, then *c s* would have been the correct amount of lap. But

we commenced this construction with a travel that we knew to be too great; hence, to find the correct travel and lap, we must proceed as follows: Join the points B and y. From s towards B, lay off on the line $A\,B$ a point b; the distance between the point s and b must be equal to the width of the steam port plus the amount that the valve is to travel beyond the steam port, which, in this example, is assumed to be $\frac{1}{8}$ of an inch. Therefore the distance from s to b must be $2\frac{1}{8}$ inches. Through b draw a straight line $b\ y_2$ parallel to $B\ y$, intersecting the line $y\,g$ in the point y_2. Through the point y_2 draw a straight line $y_2\,x_2$, parallel to the line $y\,x$, and intersecting the line $A\,B$ in the point c_2. From c_2 as a center, and with a radius equal to $c_2\,b$, or $c_2\,y_2$, describe a circle $a\,b\,y_2$. Then $a\,b$ will be the travel of the valve, which, in this case, is $7\frac{5}{8}$ inches, and the distance from c_2 to s will be the lap, which, in this example is 1-11/16 inches.

The Events of the Distribution of Steam.

The outside edges c_2 and c_3 of the valve, and the outside edges o and o_2 of the steam ports, will regulate the admission and suppression of steam; the inner edges i and i_2 of the valve and the inner edges s and s_2 of the steam ports control the release and compression of steam. The parts of the stroke of the piston during which these events will happen can be found by the following methods:

Example.—Travel of valve, 5 inches; lap, 1 inch; lead, ¼ of an inch; stroke of piston, 24 inches; no inside lap or clearance. Find at what point of the stroke the admission, suppression, release, and compression will take place.

In Figs. 30, 31 and 32 the valve occupies different positions, but the sections of the valve in these figures are exactly alike, because they represent one and the same

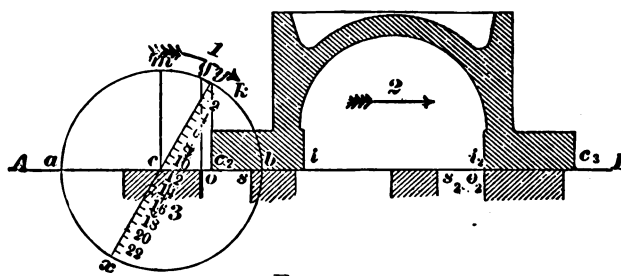


FIG. 30.

valve. In Fig. 30 the distance between the edge c_2 of the valve and the edge o of the steam port is ¼ inch, which is the amount of lead given in our example; hence, this position of the valve indicates that the piston is at the beginning of its stroke, and the angle $m\,c\,y$ is the angular advance of the eccentric. In Fig. 31 the edge c_2 of the valve and the edge o of the steam port coincide, and, since the valve is moving in the direction indicated by arrow 2, the suppression commences, or, in other words, the valve is cutting off steam when it is in the position as here shown. In Fig. 32 the inside edge i of

the valve coincides with the inner edge *s* of the steam port, and, since the valve is moving in the direction indicated by arrow 2, the release must commence when the valve arrives in the position here shown.

In Figs. 30, 31 and 32 the distances from the outside edge *o* of the steam port to the center *c* of the circle

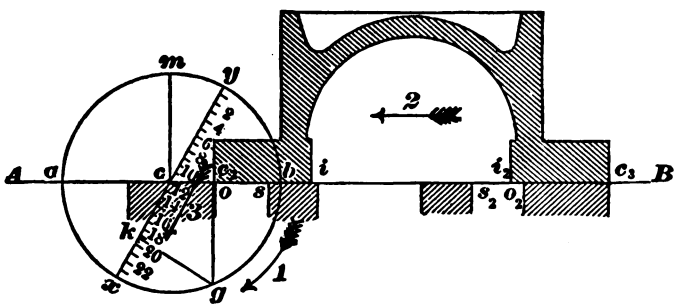


FIG. 31.

a b m are equal; that is, the points *c* and *o* are one inch apart, which is the amount of lap. The diameters of the circles *a b m* are all five inches, which is the travel of the valve given in the example, and the circumference of each circle represents the path of the eccentric, and also the path of the center of the crank-pin. The point *y* in these figures represents the position of the center of eccentric when the piston is at the beginning of its stroke. The distance between the point *y* and *m* is the same in all figures, and consequently the angles formed

by the lines $y\,x$ and $m\,c$ are equal and represent the angular advance of the eccentric.

When the valve occupies the position as represented in Fig. 30, the center line of crank will coincide with the line $A\,B$; and since the piston will then be at the beginning of its stroke, it follows that the line $A\,B$ will indicate the direction in which the piston must move. In

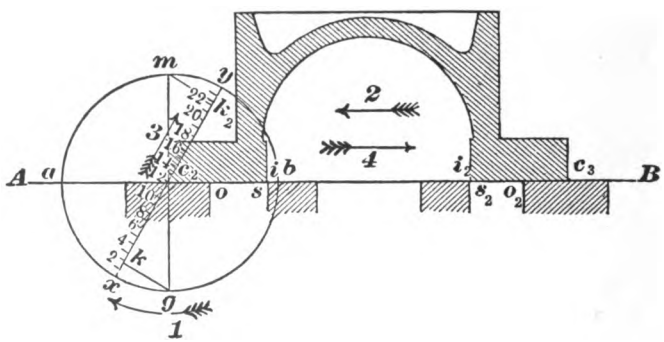


FIG. 32.

order to compare the relative position of the piston with that of the valve with as little labor as possible, we shall assume that the direction in which the piston moves is represented by the line $y\,x$, instead of the line $A\,B$; hence the point y will not only show the position of the center of the eccentric, but it will also indicate the position of the center of the crank-pin when the piston is at the commencement of its stroke. If these remarks are thoroughly understood, there will be no difficulty in comprehending that which is to follow.

Now let us trace the motions of the valve and piston and thus determine at what part of the stroke the events (previously named) will take place. When the crank-pin is moving in the direction as indicated by the arrow marked 1, Fig. 30, the center of eccentric will move through part of the circumference, *a b m*, and the valve will travel in the direction indicated by the arrow 2, thus opening the steam port wider and wider until the end *b* of the travel is reached; then the valve will commence to return, and as it moves toward the center *c*, the steam port gradually closes, until the valves reaches the position as shown in Fig. 31; then the steam port will be closed and steam cut off. To find the position of the piston when the valve is cutting off steam, we draw through the edge c_2 of the valve, Fig. 31, a straight line $c_2 g$, perpendicular to *A B*, intersecting the circumference *a b m* in the point *g;* through this point draw a line perpendicular to *y x* intersecting the latter in the point *k*, and this point *k* being 19⅛ inches from *y* indicates that the piston has traveled 19⅛ inches from the beginning of its stroke before the steam is cut off, and that steam has been admitted into the cylinder during the time the piston traveled from *y* to *k*. As the piston continues to move toward the end *x* of the stroke, the valve will move in the direction of the arrow 2, Fig. 31, and the steam port will remain closed so that no steam can enter the

cylinder or escape from it; hence the steam that is now confined in the cylinder must push the piston ahead by its expansive force, but the moment that the valve reaches the position as shown in Fig. 32 the release of steam will commence. To find the corresponding position of piston we draw through the edge c_3 of the valve, Fig. 32, a line $c_3\ g$, perpendicular to $A\ B$ intersecting the circumference $a\ b\ m$ in the point g. Through this point draw a line $g\ k$ perpendicular to $y\ x$, intersecting the latter in the point k, and this point k being 22⅜ inches from the beginning of the stroke indicates that the piston has traveled through this distance when the release of steam commences. Now notice, the steam is cut off when the piston has traveled 19⅛ inches, and the release of steam commences when the piston has traveled 22⅜ inches, consequently the steam is worked expansively during the time the piston moves 3¼ inches of its stroke. The steam port will remain open to the action of the exhaust during the time the piston completes its stroke and moves through a portion of its return stroke. In the meantime the valve will move to the end a of the travel and return as indicated by arrow 4, and the moment that the valve again reaches the position shown in Fig. 32, the release of steam will be stopped. To find the corresponding position of the piston, draw through the edge c_2 of the valve, Fig. 32, a straight line $c_2\ m$ perpendicular to $A\ B$, inter-

secting the circumference *a b m* in point *m*. Through this point draw a straight line *m* k_2 perpendicular to *y x*, and intersecting the latter in the point k_2. Since the distance between the points *x* and k_2 is 22⅜ inches, it follows that the piston has moved through 22⅜ inches of its return stroke, by the time that the release of steam will cease. As the valve continues its travel in the direction of arrow 4, Fig. 32, the steam port will remain closed until the edge c_2 of the valve coincides with the outer edge *o* of the steam port, and during this time the steam which remained in the cylinder is compressed, but as soon as the edge c_2 of the valve passes beyond the steam port edge *o*, the admission of steam into the cylinder will commence. To find the corresponding position of the piston, draw through the outer edge *o* of the steam port, Fig. 30, a straight line *o g* perpendicular to *A B*, and intersecting the circumference *a b m* in the point *g;* through this point draw a line *g k* perpendicular to *y x*, intersecting the latter in the point *k*, and since the distance between the points *x* and *k* is 23⅞ inches, we conclude that the piston has moved through 23⅞ inches of its return stroke before the admission of steam will begin. Here we see that steam will be admitted into the cylinder before the return stroke of the piston is completed, and that is the object of lead, as has been stated before. Notice once more: the compression of

steam will commence when the piston has traveled 22⅜ inches of its return stroke, and will cease when the piston has traveled 23⅞ inches of its return stroke, hence the steam is compressed during the time that the piston travels through 1½ inches.

In each of these figures the point *g* represents the relative position of the center of eccentric to that of the valve. The point *g* will always be found in the circumference *a b m* and in a straight line c_2 *g* drawn perpendicular to *A B*, the former passing through the outer edge c_2 of the valve.

The reason why the point *g* should in all cases be found in the straight line c_2 *g* drawn through the outside edge c_2 of the valve is this: the center *c* of the circle *a b m* has been placed on the line *A B* in such a position (as shown in these figures), that the distance between the center *c* and the outside edge *o* of the steam port is equal to the lap, therefore the center *g* of the eccentric and outer edge c_2 of the valve will always lie in the same straight line drawn perpendicularly to *A B*. If the distance between *c* and the outer edge *o* of the steam port is greater or less than the lap, then the center of the eccentric and outside edge of the valve will not lie in the same straight line drawn perpendicular to the line *A B*. Here, then, we can conceive the necessity of placing the center *c* of the circle *a b m* in the position

as shown in these figures. The correctness of these remarks must be evident to the reader if the explanations

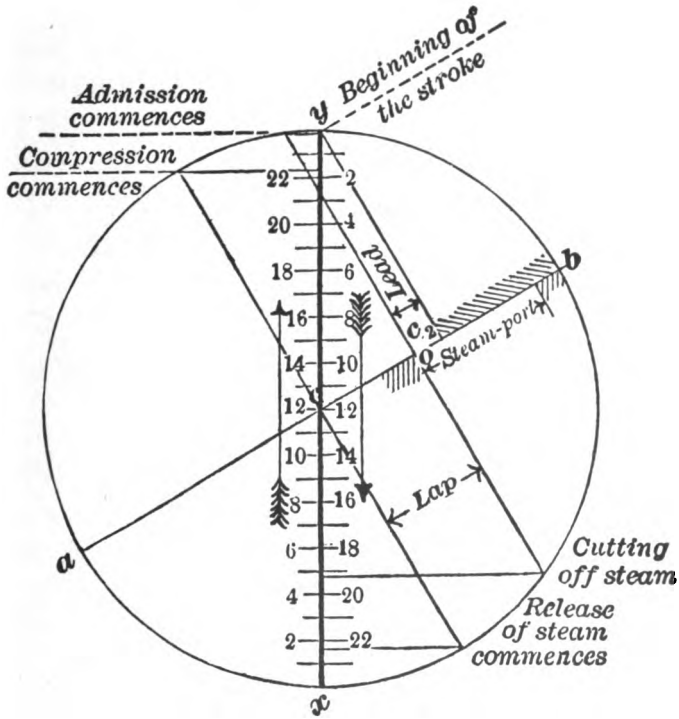


FIG. 33.

in the previous examples have been understood. Again, since we have assumed that the point *g* not only represents the center of the eccentric, but also the center of the crank-pin, it follows that, in order to determine how

far the piston has moved from the beginning y of its stroke when the crank-pin is at g, we must draw a straight line through the point g perpendicular to $y\ x$, as has been done in these figures.

From these constructions we can obtain our answer to the *example* illustrated by Fig. 31, namely:

Steam will be cut off, or, in other words, suppression will commence when the piston has traveled 19⅛ inches from the beginning of its stroke, and steam will be admitted into the cylinder during the time that the piston travels through this distance. The steam will be released when the piston has traveled 22⅜ inches from the beginning of its stroke, consequently the steam will be worked expansively during the time the piston travels through 3¼ inches. The release of steam will continue until the compression commences, which will occur when the piston has traveled 22⅜ inches of its return stroke. The compression will cease, and the admission of steam commence, when the piston has traveled 23⅞ inches of its return stroke.

The same answer to our example could have been obtained with less labor by a construction as shown in Fig. 33, which is nothing else but a combination of the three preceding figures; the methods of finding the different points in Fig. 33 have not been changed, and there-

fore an explanation in connection with this figure is unnecessary.

The Allen Slide Valve.

The Allen ported valve, which is sometimes referred to as the Trick valve, is, in general design, similar to the plain D slide valve, with the exception that it has a

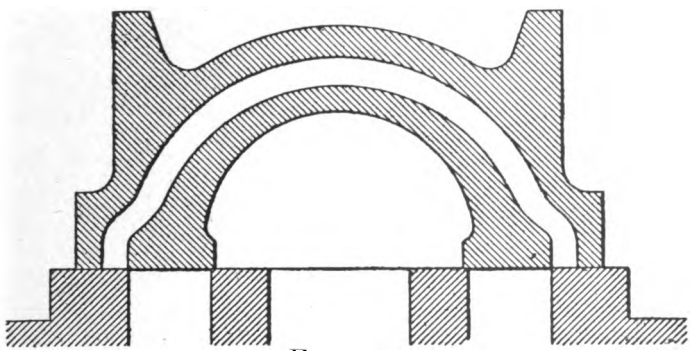


FIG. 34.

supplementary port, or passageway, cast into it, which passes over and through the crown forming the exhaust arch which ends in the two ports in the valve face, as shown in Fig. 34.

The Allen valve was designed to overcome the defects of the plain slide valve, which does not permit the use of full boiler or steam chest pressure at the beginning of a stroke, when it is most needed, without giving excessive lead, which would produce a premature cut-off and otherwise impair the operation of the valve. The Allen valve

was also designed to give a larger area for the admission of steam, and for a portion of the valve travel the area of steam port opening to the cylinder is, in fact, doubled.

The valve and seat are so arranged that they give two openings to steam for the same amount of movement necessary for a plain slide valve to give one opening; that is, there is the same amount of opening for live steam with one-half the travel of a plain slide valve. As soon as the outside edge of the valve begins to open the steam port, the supplementary passage also begins to receive steam, thereby giving a double opening for the admission of steam. As the travel of the valve is always short, when an engine is running at high speed, the advantage of the double opening is evident, because it admits the steam at the beginning of the stroke, and maintains a full pressure on the piston till the point of cut-off.

Fig. 34 shows the valve in central position upon the valve seat, and, the reader will observe, both steam ports are completely closed, the same as with the plain slide valve. Therefore the points of release or of compression will not be affected unless the lead is changed, when they will take place earlier or later in the stroke.

Fig. 35 shows the same valve moved off of its central position and the reader may note that one of the steam ports is receiving steam from each side of the valve at the same time. It may also be observed that the steam

edge of the valve and edge of the supplementary port open simultaneously, and, therefore, must cut-off at the same time.

The chief advantage derived from the use of this valve is at high speed when the engine is worked at a short cut-off, the advantage secured results from the increased

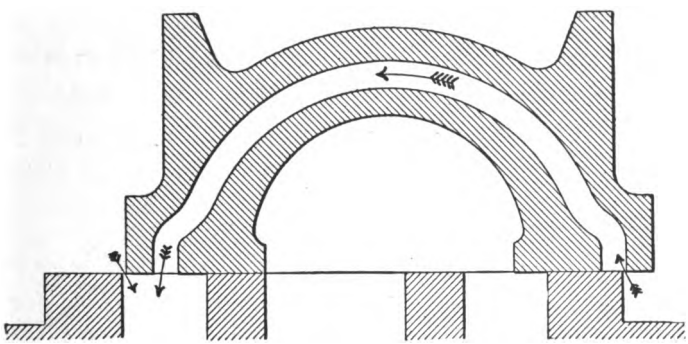


FIG. 35.

admission of steam into the cylinder, thus enabling the engine to develop increased power.

It has been claimed by many men in positions to know the facts that a saving of coal amounting to about seven per cent is obtained by the use of this valve, while others in just as good positions to know do not consider that this valve gains anything along economical lines.

One undesirable feature of the valve is the fact that when operated at short cut-offs there is double the port

area for admission that there is for exhaust—because the auxiliary port is used only for admission.

While the value of this valve has always been more or less a debatable question, it has often been condemned where the evil results could more properly be attributed to its improper adjustment or application.

The best results can only be secured from the use of this valve by designing the ports and bridges so they will exceed the full travel of the valve by at least ⅛ inch. It may also be said that full advantage cannot be obtained if the valve is given as much lead as a plain slide valve.

The rule providing that lead should be reduced in proportion to the length of the port must be modified for the Allen valve, and the lead must be further reduced to about one-half that given to plain slide valves. It may also be noted here that the Allen valve should be given enough exhaust clearance to permit a proper exhaust period, due to the fact that at short cut-offs there is double the port area for admission that there is for the exhaust, and without exhaust clearance the compression may become excessive.

The original design of the Allen valve proved impracticable when used as a slide valve for locomotives, until it was balanced, for the excessive pressure bearing

against its frail shell-like form caused a springing of the valve face and rapid wear of the valve and its seat.

We will have occasion to refer to this valve again when we reach the subject of Balanced valves.

The Miller Slide Valve.

The Miller slide valve was designed to provide a valve which would automatically position itself on its seat with respect to the ports.

It will be observed by referring to Fig. 36, that conical stops are provided for the movement of the valve on its rod, and, as a result, when the valve travels a predetermined distance up the bevel, or incline, of the cone, upon the abrupt stop of the rod to begin its return stroke, it acts as a cushion for the valve to receive the impact resiliently, thus preventing injury to the flanges at the valve ends, and also preventing the stripping of the threads on the valve rod. Another feature of the valve is that of balancing or suspending it from the cones, in order to prevent it from dragging on its seat, whereby it will neither cock, stick nor blow.

The valve is of the D slide valve type, with a longitudinal bore *a* through the valve, the ends of which are outwardly flared, as shown at *b*. A valve rod *c* extends through the bore *a* and a little beyond the outer end of the valve; and is screw-threaded, as shown at *d*, for a length exceeding that of the valve.

Threaded on the portion *d* of the valve rod *c*, at the inner ends of the valve, are cones *e* and *f*, having integral flanges *g* and *h* thereon, beyond which are angular nuts *i* and *m*, also integral with the cones, and jam-nuts *o* and *p* on the rod, adapted to be jammed against the angular portions *i* and *m* of the cone *e*. It may also be seen from the illustration that the inner diameter of the cones approximate the diameter of the bore *a*, the

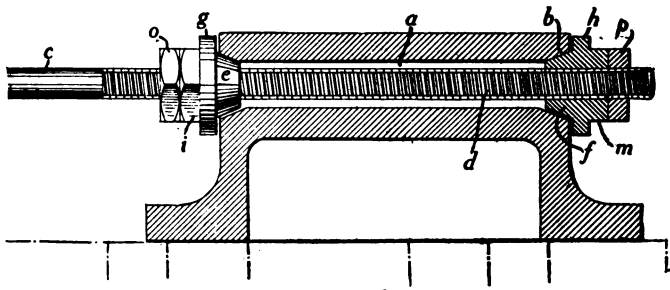


FIG. 36.

cones being beveled at the same angle as the bevel of the valve. This valve was patented April 20, 1915, by Mr. John W. Miller, of Rantoul, Ill.

Power Required to Move a Valve.

To determine the power required to move a valve, multiply the area of the valve face by the steam pressure upon it, and then deduct one-third for back pressure from steam port and exhaust port. Friction between two smooth surfaces well lubricated varies from 1/10

to 1/14 of the pressure; the weight of the valve itself being so slight that it need not be considered.

As an illustration, let us refer to a valve 10 ins. by 20 ins. with 165 lbs. of pressure per sq. in. Then $10\times20=200\times165=33{,}000$ from which deduct one-third, and the result is 22,000 lbs. This amount, if divided by 10, will be 2,200 lbs.; the amount of power in pounds which is required to move the valve. This strain on the valve gear will cause it to wear rapidly. This amount is decreased by the proportions of the reverse lever, in proportion to the distance between the lower connection of the reverse lever and the point at which the reach rod is connected to the lever, in comparison with the distance to the end of the lever. Yet the effort required of the engineer, in moving such a valve, is very great.

BALANCED VALVES.

The plain slide valve used in the early days of locomotive construction answered all purposes then required, but with the increase in size of the locomotive, together with the increased boiler pressure, it was almost impossible for an engineer to reverse an engine, and a great deal of power was wasted in moving a valve of this kind on account of the enormous pressure on the back of the valve.

History.

We have no doubt the problem of removing the excessive pressure from the back of the valve challenged alike the skill of practical mechanics and the professional engineers of early days, and we presume many devices were constructed to accomplish the purpose, but the first patent issued by the United States Government covering a balanced balve was granted to Mr. Hiram Strait, of East Nassau, N. Y., on June 25, 1834, and the second patent was secured by Mr. John Kirkpatrick, of Baltimore, Md., on July 10, 1834, but neither invention proved to be of practical advantage.

The oldest form of balance valve, which was found to be practicable, was invented by Mr. George Richardson, of Springfield, Mass., and he secured a patent covering the same on October 28, 1862.

We are unable to say exactly when or where the Richardson balance valve was first used, but about 1870 the Boston & Albany Railroad Company tried it out on an engine and the results obtained were so satisfactory that the company applied the invention to the majority of their locomotives.

Countless other forms of balancing valves have made their appearance and have departed since, but we shall confine our discussion to the various types which have stood the test and which have been found to be practical, together with a few new inventions which may demonstrate their worth later on.

Object of Balancing.

Almost all forms of balancing slide valves are constructed on the same principle, and are intended to prevent the steam chest pressure from exerting its force on the greater portion of the back, or upper surface, of the valve; thus enabling the valve to be easily moved, without excessive friction on the seat or excessive effort to lift it against the pressure which holds it down.

A slide valve may be considered balanced when the pressure on its top is just enough in excess of the pressure under it to insure the valve remaining on its seat, but not sufficient to prevent the valve from relieving excessive pressure, or accumulation of water, in the cylinder.

In the design of the valves, the balance is equally as important as the efficient distribution of steam in the cylinders, for an improper, or defective, balance will impose enormous stress upon the valve gear and connecting rods, and greatly impair the work they are required to perform. In fact it is necessary for the designer to consider, when calculating the dimensions of stems and rods, the increased work that would be imposed upon them in the event of an accident to the balancing strips and lubricating apparatus.

Area of Balance.

The size and shape of the protected area on the top of the valve varies in different forms of balancing, but it should equal the sum of the area of one steam port, the exhaust port, and two bridges, plus eight per cent of this sum for plain valves, and plus five per cent for Allen-ported valves. The balanced area should be measured from the outside of the strips, or the bottom outside edge of the inner ring when the outer ring is raised.

It may be said generally that good results may be obtained if about 65 per cent of the opper surface of the valve is protected, and 35 per cent exposed, to counteract the upward pressure of the steam against the face of the valve, which is not at all times uniform.

Methods of Balancing.

The original method of balancing a slide valve, which was first employed by Richardson, is generally followed, and the object is accomplished by fitting four iron strips ½x1½ inches, which are called balance, or packing, strips, in suitable grooves cut in the top of the valve near its outer edges, thus forming a rectangular enclosure on top of the valve. The strips are supported on coil or semi-elliptic springs to hold them against the pressure plate which is attached to the cover of the steam chest.

A later form of balancing consists of a conical ring cut through at one point, and fitted to a taper bearing on the top of the valve. No springs are necessary to support the ring since its reaction on the taper bearing, due to its elasticity and the steam pressure, tend to lift it against the pressure plate.

Hole in Top of Balanced Valve.

One or more small holes are drilled through the top of balanced valves to permit any small volume of steam which might get on top of the valve, as a result of defective balance strips, to pass into the exhaust and thence to the atmosphere, in this manner maintaining the balance feature of the valve. This hole in the top of the valve is generally referred to as the release port.

If an escape for the steam which leaked by the balance strips was not provided, such steam would, in time, accumulate on the top of the valve, and neutralize the benefits secured from properly balanced valves.

Advantages of Balancing.

The chief advantage of a balanced slide valve is the fact that it is easily moved, without excessive friction, and that it can relieve itself of excess pressure, or the accumulation of water in the cylinder, by lifting from its seat. As a result it is unnecessary to provide means for relieving excess pressure in the cylinders.

Disadvantages of Balanced Valves.

When cylinders of large dimensions are used, with high steam pressure, the slide valve becomes unduly large for a proper length of port opening and, even when the valve is well balanced, creates an excessive amount of friction when it is moved on its seat. A slide valve when used on a very large cylinder gives undue cylinder clearance due to the increased length of the ports, and the large steam chests necessary, and causes more or less steam condensation. This probably accounts for the high water rate of engines with very large valves and steam chests.

Another objection often advanced is that the end strips cut into the balance plate, and therefore, it is uncom-

mon for a locomotive to run more than 25,000 miles before the valves require refacing. This difficulty may, however, be overcome by dispensing with springs under the front and back strips, by allowing the end strips to overlap and receive their support from the long side strips. When such an arrangement is used shoulders should be provided on the bottom of the side strips so that they cannot work out of their proper positions.

While slide valves have been operated successfully with a high degree of superheat, they are not as reliable as the piston valve under high temperature steam.

The Richardson Balanced Slide Valve.

The Richardson balanced valve, briefly referred to heretofore, is similar to the plain slide-valve, but it has a certain amount of space on the top of the valve enclosed by four rectangular packing, or balance, strips, ½x1½ inches, of cast iron, which are held tightly against the pressure plate by the steam pressure, and by semi-elliptic springs placed in the grooves beneath them. The two shorter packing strips have gib-shaped ends to retain them in their proper positions, and are arranged so that no packing strip of the balance device will travel over, or rub, the path of travel of any other strips, thereby insuring a steam tight joint at various travels. The balance strips are marked *p p* on the illustration, Fig. 37.

There is also provided a pressure plate, A, which is bolted to the inside of the cover of the steam chest, but this may be cast in a single piece. The steam acts against the outside of the strips, making them steam tight, and, as a result, excludes steam from the top of the valve. A small amount of clearance between the valve and the pressure plate is allowed, so that excessive

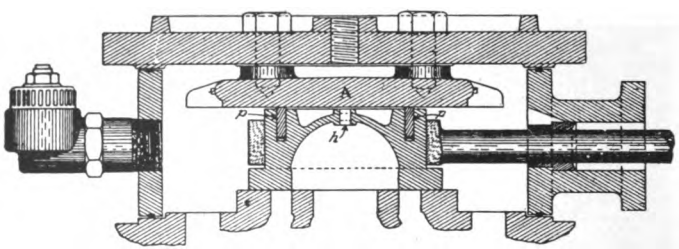


FIG. 37.

cylinder pressure may be relieved by allowing the valve to lift slightly from its seat. A small hole, *h*, is drilled through the top of the valve, so that any steam that escapes past the packing strips may pass out to the exhaust without exerting pressure on the top of the valve.

The proper area to be enclosed within the four balance strips may be determined by adding the area of the exhaust cavity to the area of one steam port. If a greater area be enclosed there is danger of overbalancing the valve, and a smaller area will often give satisfactory results.

The reliability of this type of balanced valve, together with its simplicity of construction, and the fact that the balance feature can be used with various forms of slide-valves, have resulted in its general adoption, and it is now used extensively.

The Allen-Richardson Balanced Valve.

This well known type of balanced valve is simply a combination of the Allen ported valve and the Richard-

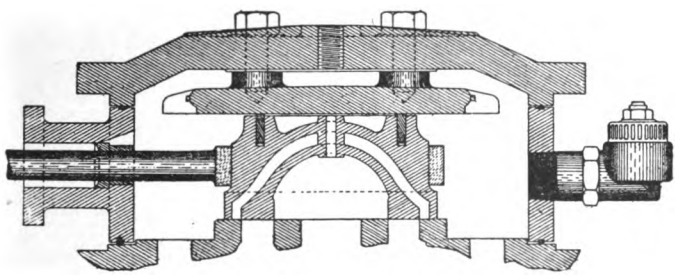


FIG. 38.

son system of balancing, which is shown in the sectional view presented in Fig. 38.

We pointed out in the preceding pages, when analyzing the Allen valve, that it was not practical for locomotive work, until it was balanced, but when used in connection with the Richardson system of balancing it gives very good results, and thousands of this combination type are now in service.

American Balance Valve.

This form of balance valve is used quite extensively in locomotive service, and its popularity is, no doubt, due to its simplicity of construction, positive action, economy of maintenance, and large area of balance.

The valve is balanced by a single or double cone, or disk, cast on the back of the valve, the single cone being

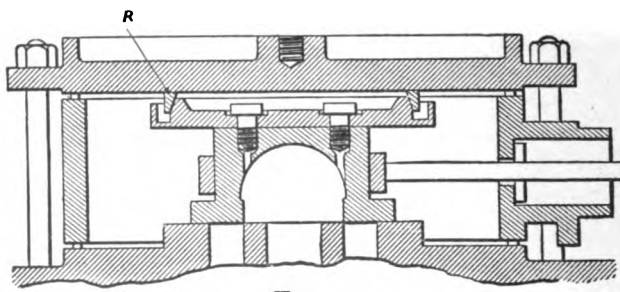


FIG. 39.

preferable where steam chest space will permit its use. These cones have a beveled face, are fitted on the outside by taper rings, marked R in the illustration, Fig. 39, of the same bevel as the cone rings, made ¼ inch smaller than the bevel of the cone, and are expanded over the cone so that it does not require a spring.

The conical ring is cut through at one point, in order that the ring may expand, and the opening is protected by an L-shaped plate of the same depth as the ring, which fits against the cone and insures a steam tight joint.

Owing to the natural elasticity of the ring and its expansion over the cone, a tension is placed on the ring, the function of which is to close the ring on the cone, which necessarily moves upward. The ring is therefore self-supporting and self-adjusting.

For the valve to lift from its seat it is necessary to force the cone up into the taper ring; and since the ring is held, by the steam chest pressure, from opening, the valve cannot lift without first overcoming the entire friction of the beveled face, besides opening the ring against the steam chest pressure. Of course, if the taper was made large enough, say 45 degrees, the action of the steam chest pressure on the circumference of the ring would wedge it between the cone and the steam chest cover and, as a result, exert an enormous pressure on the valve, so that it would not work satisfactorily. Extensive experiments, however, have been made, with the taper from 9 to 24 degrees, and, from actual tests, the proper degree of taper has been ascertained whereby the ring will rise under all conditions without crowding the balance plate more than is necessary to give good results.

The rings are made of hard close-grained cast iron from standard gauges, which are fitted on the lathe in place of a caliper, or rule, and as they are made 1 inch deep, they can easily wear 3/8 inch and still adjust them-

selves. As the ring is turned 1/4 inch smaller than its working diameter, it can be readily seen that 1/32 inch in the diameter of the ring either way from the size specified would not in any wise interfere with the service of the valve.

The rings are all bored smaller than the diameter at which they are to work; therefore when a ring is set on its cone it will stand higher than in its working position. The face of the pressure plate must not be closer than 1/8 inch to the top of the cone after the steam chest cover has been screwed down, for, in placing the steam chest cover in this position, the ring is expanded over the cone until its inside diameter at the bottom is the proper balancing diameter.

Sufficient allowance must be made for the removal of the valve-yoke, and, as a result, a detachable disk is used, but the makers recommend an arrangement whereby the valve-yoke may be carried on the steam chest at the ends of the valve.

When the valve is in position, and the steam chest cover has been bolted down, there must be 1/8 inch space between the face of the upper bearing (in this case the pressure plate), and the top of the disk, or cone. This arrangement will allow the valve to lift from its seat 1/8 inch, which it will do as soon as steam is shut off

while the engine is in motion, if it is not held down by the valve-yoke.

A casual inspection of the illustration may lead the reader to believe this valve is overbalanced, but the impression will disappear when proper consideration is made for the reaction from the surface of the bevel ring, and friction of the conical joint.

This design of balance valve was invented May 3, 1892, by Mr. W. J. Thomas, of Sausilito, Cal., and the improved rings were patented June 21, 1898, by Mr. John T. Wilson, of Jersey Shore, Pa. The valve is manufactured by The American Balance Valve Company of Jersey Shore, Pa.

Gould Balance Valve.

The Gould balance valve was designed to overcome the tendency of balance valves, when slightly worn, to suffer from steam leakage; and also with the idea in view of obtaining a valve that will take up its own wear, so that this difficulty is very largely overcome.

This valve is constructed like the plain D slide valve, with the same face measurements, and it is intended to work on the same ports and has the same admission, cut-off and release as the common D slide valve. Fig. 40 shows this valve with cover displaced and rings raised.

This valve is semi-circular or half round in shape, and the flat portion travels on the valve seat; while in operation it travels under a half round, or semi-circular, pressure plate which rests on the valve seat; it has expansion half rings, four in all, two at each end of the valve, and these rings expand, with steam, against the pressure

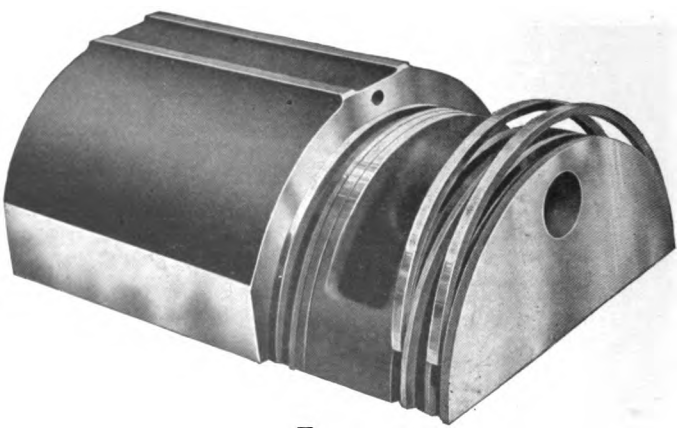


FIG. 40.

plate, thus preventing steam from entering or escaping at each end of the valve.

Between these rings at each end is a groove of exactly the same length as the packing grooves and it extends around the oval portion of the valve; when the rings are expanded this groove becomes a steam chamber, as steam is taken from the ports in passing over them, through holes bored in the face of the valve to this

chamber or circular groove. The semi-circular pressure plate keeps the steam from pressing downward upon the back of the valve proper; the valve is balanced against port pressure by steam taken from the ports through to the chamber previously described; the pressure plate is kept from moving endwise by pins in each end of the plate which rest against the walls of the steam chest,

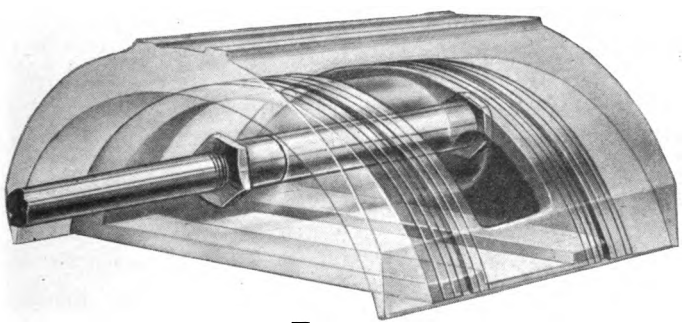


FIG. 41.

and is held down firmly by a set screw through the steam chest cover.

An X-ray view of the valve through its semi-circular casing, showing its interior construction, and the sleeve, or valve rod connection, is presented in Fig. 41.

The advantage of the Gould valve is that there is no boiler pressure on the back of the valve, as the cover fully protects the valve from live steam, and the rings exclude the steam from passing beyond the ends of the valve, as there is no opening for steam to get between

the cover and the valve. The ends of the valve are exposed to live steam; but, the pressure on both ends being equal, there is no end friction. The communicating ports always allow the same pressure on the back of the valve that is on the face of the valve, thus keeping it perfectly balanced at all times. This valve is manufactured by the Gould Balance Valve Company, of Kellogg, Iowa.

The Mewes Valve.

Several novel features are presented in a valve patented May 22, 1917, by Mr. Richard W. Mewes, of Des Moines, Iowa, so we believe a detailed description of the same will be of interest to the reader.

After stating that the object of the invention is to provide a valve of simple, durable and inexpensive construction, for engines, particularly locomotives, the inventor claims the desired result is attained in the construction and arrangement of the various parts of the device, as shown in the accompanying illustrations, which he describes as follows:

Fig. 42 shows a top, or plan view of the steam chest with my improved valve therein, the top or cover of the steam chest being removed.

Fig. 43 shows a vertical, sectional view, taken on the line 2—2 of Fig. 42.

Fig. 44 shows a top, or plan view of one end of my valve with the cover plates 34 and the plate 39 removed

for the purpose of illustrating other parts of the invention.

Fig. 45 is a detail, perspective view of part of the packing and balancing mechanism at the upper part of the valve.

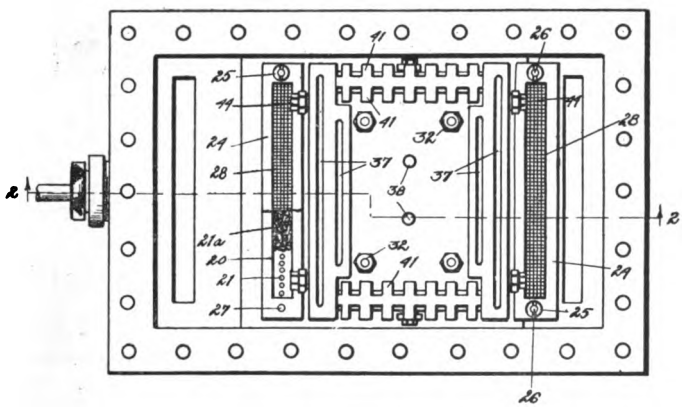


FIG. 42.

Fig. 46 is a front elevation of a portion of the front end of the mechanism shown in Fig. 45, showing the manner in which the two packing plates or strips are arranged at right angles to each other and joined, and

Fig. 47 shows a vertical, sectional view taken on the same line as Fig. 43, but enlarged for the purpose of clearness, and showing only a small part of the valve.

In the accompanying drawings, I have used the reference numeral 10 to indicate generally the steam chest of

a locomotive engine, having the intake ports 11 arranged in the bottom of the steam chest near the ends thereof, and having the main exhaust port 12 preferably in the center of the bottom of the steam chest and having the ports 13 arranged on opposite sides of the port 12 for steam from the opposite ends of the cylinder.

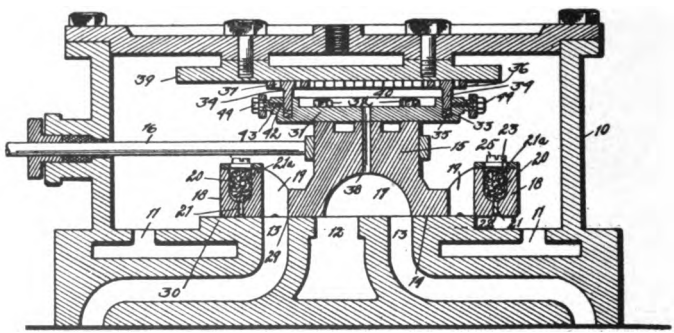


FIG. 43.

The portion of the bottom of the steam chest forms a valve seat 14. Arranged to slide on the seat 14 is a sliding valve, having a main central body portion 15, operatively connected with which is the valve stem 16, slidably extended through the side of the steam chest. In the lower central portion of the body 15 of the valve is a recess 17 adapted at all times to be in communication with the port 12, and adapted in different positions of the sliding movement of the valve also to be selectively in communication with the ports 13.

Formed on the lower portion of the valve body are opposite lateral extensions 18, which, at their outer ends, are thicker than the intermediate portions of said extensions 18, as shown in Fig. 43. Formed in the extensions 18 between their ends and the body 15 are ports 19.

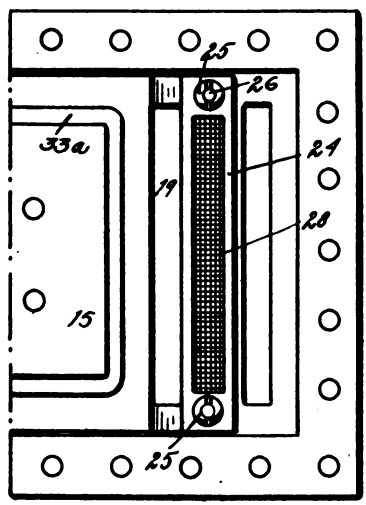


FIG. 44.

The right-hand port 19 on the left-hand movement of the valve is adapted to register with the right-hand port 13, and the left-hand port 19 is adapted in the right-hand movement of the valve to register with the left-hand port 13.

In the upper portions of the outer ends of the extensions 18 are recesses 20 in which I preferably mount

suitable material 21ª, such as waste or the like, for collecting, holding and distributing a lubricant.

Extending downwardly from the lower parts of the recesses 20 is a plurality of small passages 21 whereby a lubricant can pass downwardly from the recesses 20 to the lower surfaces of the extensions 18. The lower surfaces of the extension 18 are provided with grooves

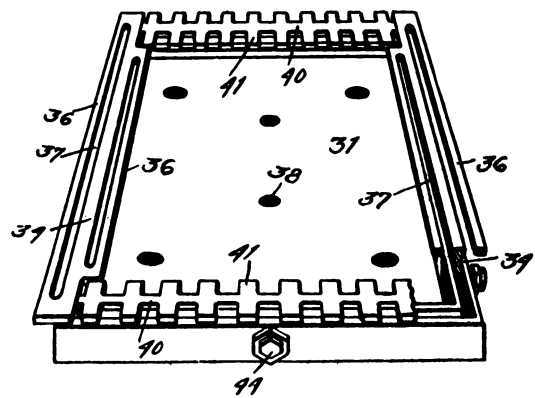


FIG. 45.

22, for connecting the lower ends of the small passages 21.

The upper portion of the outer ends of the extensions 18 are preferably covered with a screen or the like, 23. The recesses 20 do not extend to the ends of the extensions 18.

Resting above the screen are frames, 24, adapted to be secured to the extensions 18 by means of screw-threaded

bolts 25, having in them openings 26 registering with the opening 27 extending downwardly through the ends of the extensions 18, whereby a lubricant can pass through the screws downwardly through the extensions 18. The central portions of the frames 24 are cut away at 28 to admit the lubricant from the steam chest to the screen 23, and therethrough to the material 21^a and the passages 21.

Before describing the structural features of my improved valve whereby it is balanced, I will explain the operation and advantages of the extensions 18.

D valves of the type used in locomotive engines are ordinarily constructed without such extensions, and with portions which extend as far as the inner walls of the passages 19. It will thus be seen that in the ordinary D valve there are no passages 19, but steam in the different positions of the valve passes downwardly past the sides of the D valve into the ports 13. It is well known that in D valves there is a pressure, at certain times, upwardly on the sides of the valve, so that the valve tends to have a rocking motion, whereby the corners of the valve at 29 tend to become worn and rounded, and therefore the valve is not tight, and leaks steam. This is an objectionable feature of the ordinary D valve, which is well known, and it is one of the principal objects of my invention to construct a D valve in

such form as to overcome this objection by preventing the rocking movement and by taking off the wear of the corners 29. This purpose is partially accomplished by the construction hereinbefore described, and partially accomplished by the construction whereby the valve

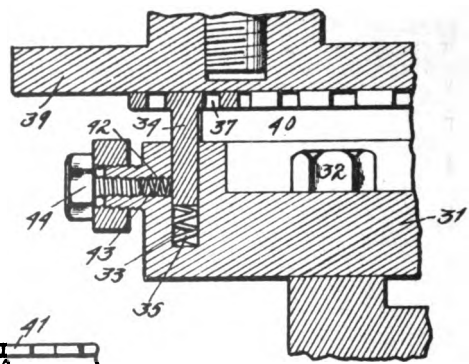


FIG. 46.

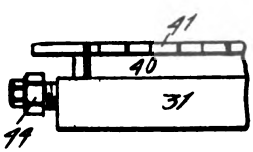


FIG. 47.

is balanced as will be hereinafter more fully explained.

By the construction already described, and by the use of the extensions 18, it will be seen that the lower surfaces of the extensions 18 beyond the passages 19 engage the valve seat, so that there is no tendency for the valve to rock, and the wear will be upon the corners, or edges, 30, which will not cause any leakage or loss of steam. The corners or edges 29 will remain sharp, so that the valve will retain its efficiency.

In order to secure complete and perfect lubrication for the sliding parts of the valve and valve seat, I have provided the means above described, including the material 20, so that the oil will pass downwardly through the recess 20 and the passages 21 to the surface of the valve seat, where it will be distributed along the groove 22, and perfect lubrication between the extensions 18 and the valve seat will be maintained.

The accomplishment of such lubrication I consider as another very important advantage of my improved valve.

I will now describe the construction whereby my valve is balanced. Resting upon the upper part of the valve is a plate 31, which, as illustrated in Fig. 43, is of considerably greater width than the body portion of the valve. In this connection it may be stated that the area of the projecting portions of the plate 31, including the area of the body 15 above the recess 17, is substantially equal to the area of the valve which is subjected to the downward pressure of steam, so that when the valve is in operation, the upward pressure of steam will tend to balance the downward pressure thereof on the valve to balance the valve. The plate 31 is secured to the body 15 of the valve by means of bolts 32.

Near its side edges the plate 31 is provided with longitudinally arranged grooves 33. Mounted in the grooves

33 are upright plates 34 which may slide vertically in the grooves 33 and are yieldingly held at the upper limit of their movement by means of springs 35 in the bottoms of the grooves 33.

At the upper end of each plate 34 are opposite lateral extensions 36, clearly shown in Fig. 43, and it will thus be seen that the plates 34 with the extensions 36 have substantially the form of a T-iron.

In the extensions 36, on opposite sides of the plate 34, are longitudinal slots, or openings, 37. The use of the extensions 36 also gives a large bearing surface at the upper end of each plate 34, so that there will be less tendency for the plates 34 to rock and wear against the sides or walls of the grooves 33.

Extended through the body 15 and through the central portion of the plate 31 are passages 38, whereby any steam which may leak past the plates 36 may pass downwardly to the exhaust port 12. The openings 37 afford passages for a lubricant whereby the lower surface of the balance plate 39, against which the upper part of the plates 34 and the extension 36 slide, may be thoroughly lubricated.

In the upper surface of the plate 31 are grooves 33^a, similar to the grooves 33, arranged at right angles thereto near the outer edges of the plate 31, and in the grooves last described are mounted plates 40, similar to the plates

34, which at their ends fit against the plates 34, the inner extensions 36 being cut away at their ends to allow the ends of the plates 40 to bear snugly against the plates 34. At their upper ends, the plates 40 are provided with a plurality of opposite, laterally extending tongues 41, which afford broad bearing surfaces, and at the same time allow spaces for the admission of a lubricant to the plate 39. The plates 34 and 40 are similarly mounted in the grooves in the plate 31.

It will be understood that there is oil in the steam passing through to the valve, so that the wearing parts of the valve are lubricated by oil furnished with the steam, and held in suspension therein.

In the edges of the plate 31, on each side thereof, are horizontal openings extending from the grooves 33 and 33^a, respectively, to the outer surfaces of the plates. In the openings 42 are mounted springs 43. Outside the springs 43 are mounted screw-threaded bolts 44, whereby the springs 43 are held against the plates 34 and 40 for maintaining tight joints at all times. It will thus be seen that by means of the plate 31 my valve is balanced and at the same time the upper moving parts are thoroughly lubricated.

My improved valve, on account of the extensions 18 and the means for lubrication, will wear much longer than the ordinary sliding D valve used on locomotives.

Thus by the use of the extensions on the valve, which overlap or travel upon the parts of the valve seat which project beyond the passages 13, during all operations of the valve, two vitally important purposes are accomplished, which are two of the main purposes of this improvement. One of these purposes is the affording of the broad bearing for the lower surface of the slide valve, whereby the ends of the valve are supported at all times on the valve seat, and the rocking or tilting of the valve, which is a serious defect in many slide valves, is wholly avoided. The other purpose is to afford a receptacle for the oil receiving material 21[a] which absorbs from the steam the oil in solution therein, and furnishes lubricant which can pass downwardly through the passage 21 for thoroughly and completely oiling the valve seat.

The extensions, particularly when formed with the oiling means shown, permit the oiling of the surfaces of the valve seat which extend between the passages 13 and the extreme ends of the valve seat, by oil from the steam in the steam chest, and also by oil passing through passages 21. On each movement of the valve, the under surface of the extension is lubricated by oil in the steam, as is clearly illustrated from the showing of the right-hand extension in Fig. 43.

The valve seat is also lubricated by oil in the steam which passes downwardly through the passages 19. It

thus appears that the surfaces of the valve seat and the valve are more thoroughly and continually lubricated than could be possible with the ordinary form of valve having no such extensions.

It is well known that in ordinary locomotives a constant supply of oil is furnished to the steam so that when the steam passes to the slide valve it contains oil. Where everything is working satisfactorily oil will be fairly well distributed over the working parts of the slide valve. There are times, however, when the supply of oil to the steam may be accidentally shut off, and there are also times, as when the engine is coasting, when the slide valves are working but no steam is being furnished thereto and consequently no oil is being furnished to the valve parts. By providing the recesses 20 and the waste 21^a or other material in said recesses, for collecting, holding and distributing oil, I provide means for taking up surplus oil if there should be more than enough to properly lubricate the parts, and holding it until such time as not enough oil is furnished to saturate the waste 21^a, whereupon oil will pass from such waste as a sort of reservoir through the passages 21. I thereby obtain a more even and continuous and thorough lubrication than could otherwise be done.

It is also clear that by the use of the extensions, it is impossible for the valve and valve seat to wear convex

and concave, particularly along the lower edges of the valve, as occurs where the ordinary D valves are used, insomuch as with my valve, the valve is held properly seated at all times and the extensions prevent any rocking of the valve. Where the ordinary D valve is used, the valve and valve seat wear convex and concave, which requires the re-surfacing of the valve by chipping or the like.

In the actual use of locomotives there is, of course, a constant leakage during the time when the valve is deteriorating until it becomes so bad it must be repaired. The loss due to such leakage is avoided where my valve is used.

The Miller Double Acting Slide-Valve.

A double acting slide-valve designed to embrace the advantages derived from inside clearance, and obviate the disadvantages resulting from excessive outside lap, by extending the period of exhaust and delaying compression until substantially the point of admission, without delaying admission or hastening cut-off, was patented January 13, 1914, by Mr. Joseph Miller, of Chicago, Ill.

The principal feature of the improvement is the employment of an auxiliary valve, sliding on and above the main slide valve, alternately opening and closing the two small auxiliary ports—one at each end—which extend through the main slide valve.

In Fig. 48 it may be observed the main valve is moving to the right, and the auxiliary valve is being moved in the same direction, by special operating mechanism, and the upwardly projecting portions, or buffers, of the main

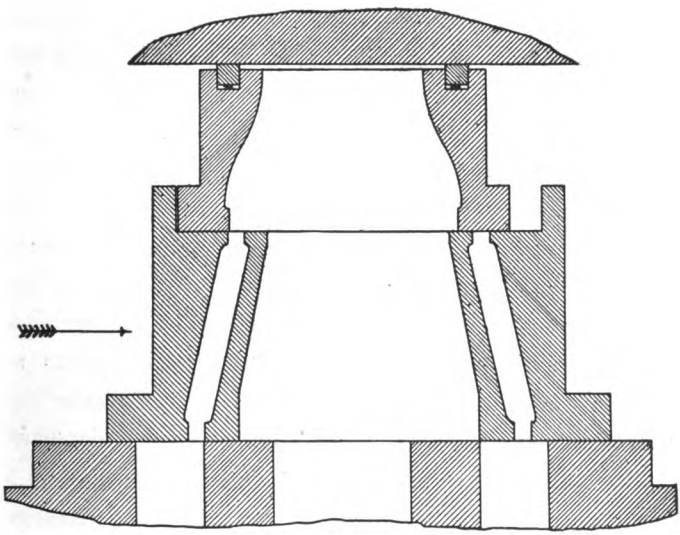


FIG. 48.

valve, against which the auxiliary valve abuts. It will also be seen by inspection that the center exhaust passage-ways extending vertically through the center of the main and auxiliary valve are always in communication.

The auxiliary exhaust passages, which extend entirely through the body of the main valve, are positioned to register with the cylinder ports in the valve seat after

the inside edges of the main valve have passed inwardly beyond the ports, and delay compression by providing means of escape for the steam which would normally be trapped in the cylinder ahead of the advancing piston.

This is manifest by an inspection of the left-hand port shown in the illustration, for the point of exhaust has been reached in the right-hand port while exhaust continues in the left-hand port; in fact, the left-hand port will remain open for exhaust until ¼ inch of the port opening for steam admission.

At the same time it will be seen that premature release in the right-hand port is prevented by the auxiliary valve closing the upper end of the vertical auxiliary passageway extending through the main valve, so as to prevent release until the inner edge of the main valve has moved to a position to open the right-hand cylinder port.

The auxiliary valve travels back and forth in the steam chest with, and in the same direction as, the main valve, as the latter approaches the end of its stroke.

The two valves are balanced in the usual manner by the use of springs, or other resilient means, interposed beneath packing strips which fit in grooves and are forced against the pressure plate, to produce a seal between the interior of the valves and the live steam compartment.

The main valve is lubricated by means of steam, trapped in a series of transverse grooves, or steam

pockets, cut in the valve face beneath the valve, as the latter slides thereover. The expansive force of the trapped steam in these pockets tends to raise the valve from its seat, thereby reducing friction and eliminating the necessity of artificial lubrication. In fact it is claimed that this system of lubrication permits the valve to be used with success on a high degree superheated engine.

It has been demonstrated that one of these double acting valves having 1 inch outside lap, and ⅜ inch exhaust lap, will give 40% increase of expansion. With 1 inch outside lap, and exhaust line and line, however, a decrease of compression of about 75% over the plain slide valve, may be obtained.

A set of these valves have been in use on an engine of the Illinois Central Railroad Company for about three years, and so far neither the valve nor the valve seats have been refaced, nor has as much as a file been touched to them. The locomotive equipped with this valve shows a good coal and water record, leading all other engines of her class in similar service by a good percentage, but an official test has never been conducted, and therefore accurate data cannot be given.

The Walsh Balanced Slide-Valve.

The valve shown in the accompanying illustration is unlike the ordinary D shaped type in its construction, for it is composed of two triangular end pieces, which

are connected by a hollow cylinder, through which the valve stem is projected.

The valve proper is enclosed entirely in a triangular hood, or covering, which is held in position by a set screw projecting through the top of the steam chest cover.

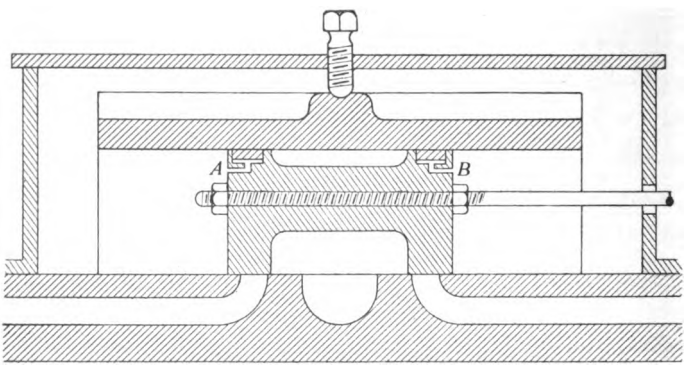


FIG. 49.

Each end member of the valve is provided with a groove in its apex, or top, extending the length of one face, and a metal balance strip of less thickness than the depth of the groove. A side view of this groove, with the balance strip set in it in the position it occupies when subject to steam chest pressure, is clearly shown in Fig. 49.

The space, or opening, below each balance strip, and between it and the bottom of the groove, is connected

by means of pin hole ports, or openings, which are marked A and B, with steam chest pressure.

In operation, live steam from the steam chest is admitted in the grooves, beneath the metal balance strips, through the small ports A and B, and the steam pressure forces the metal balance strips outward and upward, against the top and one inner face of the hood, or covering, thus equalizing the pressure of the valve on its face and balance plate. As a result the valve is well balanced, for the steam admitted below the balance strips through the ports A and B accomplishes the same results as the springs usually employed in balanced valves. In fact this method of balancing is more reliable than springs, for there is no danger of a suspension in the balance, and it will not cause the balance strip to bear against the top or inner face of the hood, or covering except when the engine is working steam.

This valve was patented June 3, 1913, by Mr. James P. Walsh, of Marengo, Iowa.

The Wilson Balanced Slide-Valve.

This valve was designed for high-pressure, and, as it is of the "Grid Iron" type, it is very light, and, being the only moving part, the reciprocating weight is the minimum.

It will be observed that the valve has two faces which are parallel and of identical construction; that the lower

surface of the balance plate is of the same area and shape as the raised valve seat, that the pockets I I are of the same area as the steam admission ports to the cylinder; that the "floating" balance plate can rock slightly and lift to relieve the cylinder when the engine

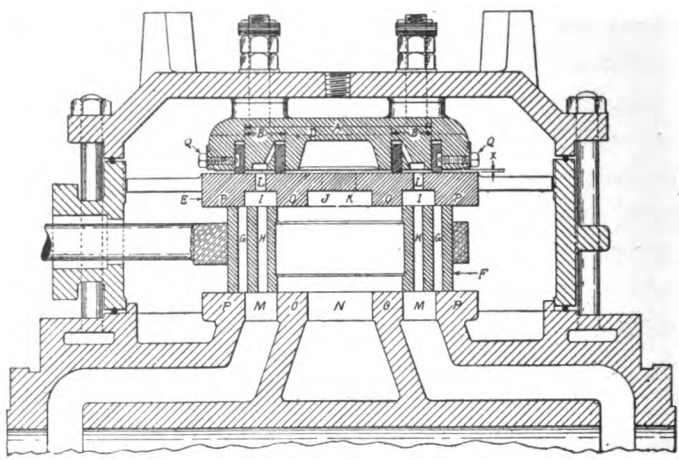


FIG. 50.

is drifting, inasmuch as the plate is located a short distance from the under surface of the pressure plate; and that the packing strips are stationary.

The inner area of the top surface of the balance plate bounded by the end packing strips and outer strips, which is to be enclosed from the pressure of steam within the chest, is determined in any given case by experiment. However, the main requirement is that the area of the

balance plate exterior of these packing strips, which is subjected to steam pressure, should be sufficient to hold the valve in place and allow it to move from its central position when moderate force is applied to the valve stem.

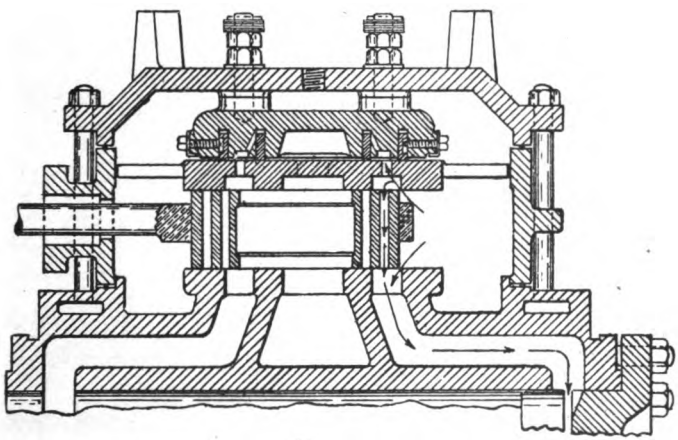


FIG. 51.

When the valve is moved from mid-position, Fig. 50, to the opening position, Fig. 51, steam passes directly downward through the admission passage M to the cylinder. At the same time steam passes to the enclosed area B at the top of the balance plate. Steam pressure upon the top and bottom faces of the valve, where it laps the port, and the top and bottom faces of the balance plate at the edge adjacent the port, is thus equalized. In the wide open position, Fig. 52, the balance is pre-

served, inasmuch as the top face of the valve at the left has passed out from under the balance plate the same distance its lower face has passed over the edge of the valve seat. When the valve covers the port, the pressure upon the top and bottom faces and upon the

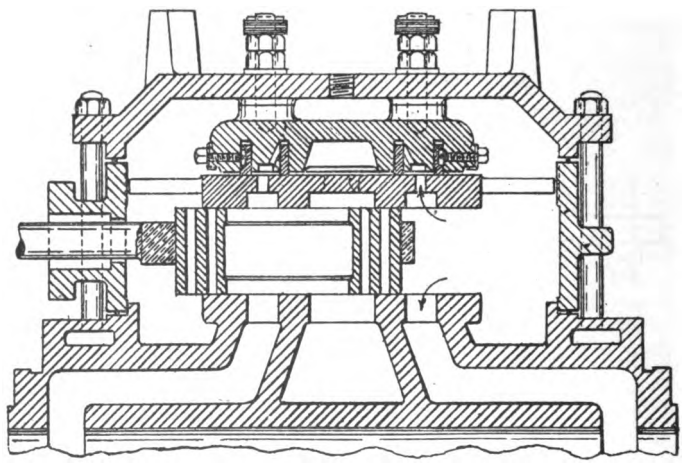


FIG. 52.

top and bottom surfaces of the balance plate is obviously constant. As the valve moves farther toward the right, and reaches the position of exhaust, Fig. 53, steam passes directly from the port, and also by way of the exhaust passages to the exhaust cavity. At the same time steam in the inclosed space between the balance strips above the port passes out into the exhaust cavity, thus preserving the balance which otherwise would be destroyed by

the top surface of the valve excluding from steam pressure the under surface of the balance plate adjacent to the pocket therein.

The balance area varies with each stroke, or movement of the valve, an amount sufficient to correspond to

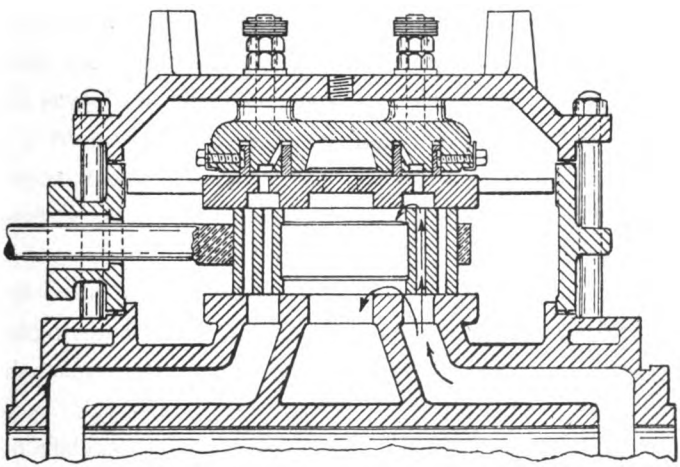


FIG. 53.

the changed position of the valve on its seat at the different points of its travel. It also provides a double exit, and admission, for the release and admission of steam.

This valve was patented March 9, 1909, by Mr. John T. Wilson, of Jersey Shore, Pa., and is manufactured by the American Balance Valve Co., of Jersey Shore, Pa.

PISTON VALVES.

The first patent issued by the United States Government covering a piston valve was secured by Mr. M. P. Mack, of Canandaigua, N. Y., in March, 1866, about 32 years after a patent was first issued for the balanced valve. We have pointed out, however, that the early balanced valves proved to be impractical, and it was not until the year 1862 that a balanced valve of merit was produced. We may, therefore, say, comparatively speaking, that the piston valve is as old as the balanced type.

The piston valve, like the balanced valve, was not generally adopted for many years after its invention, but there were more serious obstacles in its path than confronted the early balanced valves. For example, the piston valve was obliged to overcome the popularity of the balanced valve which had been before the public for some years, and had answered all requirements of the low pressure engines then in use. The piston valve was criticised because its use resulted in the rapid wear of the valve cage at the port openings, which was partly due to the absence of bridge strips in the port openings as ring bearers for the piston. Another objectionable feature was that the tallow used in early days for lubrication prevented the piston valve from giving good results.

In some of the earlier forms of piston valve the steam was admitted at the outer ends of the valve and, as a result, the action of the valve, in such cases, was precisely the same as that of the plain slide-valve. We do not believe, however, that any outside admission piston valves are now in use, with the exception of a few special types applied to existing slide valve cylinders, which will be presented later.

After some of the objectionable features of the piston type of valve had been eliminated, and various improvements had been made in its construction, the valve was, in time, adopted for marine engines, where it rendered good service. As a result, it has grown into popular favor in locomotive construction, and is now used extensively.

The piston valve is of cylindrical form, and derives its name from the fact that it consists of two pistons, one to each port, which are connected by an ordinary spindle. The valve works in a cylindrical valve seat, in which port openings are made completely around the valve.

This arrangement allows greater port area, and requires less space for the operation of the valve, for this type of valve is generally made long enough to bring the two faces, or working edges, of the valve close to the ends of the cylinder, thereby reducing the clearance space between the valve face and the piston.

This type of valve has been well described, by numerous authors, as an ordinary slide valve with its plane surface and rectangular steam ports rolled into cylindrical form, and we are willing to adopt this definition as satisfactory.

General Design.

As a general rule the piston valve is made hollow, with as large a passage through the center as possible, to secure a large area for the exhaust at the instant release occurs, so that the back pressure will be reduced to a minimum. The exhaust produces more or less pressure on the end of the valve rod and, as a result, the follower ribs of the valve should have a long bevel.

The idea that the effect of a contracted exhaust nozzle is more apparent, and detrimental, with the piston type of valve than with a slide valve, is erroneous, and may be explained by directing attention to the fact that back pressure from the exhaust acts on the outer ends of the piston valve, and, as the result of short ports, it exerts more force on the pistons.

The Built-Up Type.

The most common type of built-up piston valve is constructed of two follower plates made of cast steel or malleable iron, two cast iron skeleton or bull rings, and a single connecting piece called a spool.

The Solid Type.

As a general rule the solid type of piston valve is now made hollow, and is supplied with three light section snap rings—about ⅜ by ½-inch—on each end. From an economical standpoint the solid type is preferable, for it can be manufactured and maintained at lower cost, but in case of an accident the entire valve must be replaced, while a renewal of the damaged part is all that is necessary with the built-up type of valve.

Size of the Piston Valve.

There is a great difference of opinion regarding the proper size of the piston valve that will give the best results for large high pressure engines, but the prevailing opinion at present appears to be, that the circumference of the valve, minus the space occupied by the bridges, must, to a considerable extent, exceed the length of the slide-valve port for a similar bore of cylinder. This is accounted for by the resistance offered by the bridges, and by the fact that the portion of the port diameter opposite the cylinder cannot be so effective as that nearest it. The port ordinarily should be so proportioned that the area around the outside of the bushing at any point is equal to the combined area of all openings above this point. Formerly the use of a piston valve 55 per cent of the diameter of the cylinder was considered good prac-

tice, but of late there has been a tendency toward a smaller diameter of valve, and it has been demonstrated that a 12-inch diameter is large enough for cylinders up to 27 inches in diameter.

The size of the valve can be changed by increasing or decreasing the thickness of the valve bushing, and it is evident that a reduction in the size of the valve will reduce its weight and its stress on the valve motion, especially at high speed.

Without doubt, it is possible to use a smaller diameter of valve with superheated steam than with saturated steam, and this is probably due to the fact that the velocity of superheated steam through the ports is much greater.

The only point of similarity between the piston and slide-valve is that they are both employed to control the admission and release of steam. In all other respects they are different, but we shall here confine our examination to the difference in operation between the two.

Operation of the Piston-Valve.

The piston valve is generally of the inside admission type, with outside exhaust; that is, the inner edge of the piston valve is the steam edge, while the outer edge controls the exhaust. As a result of such arrangement all of the events of the stroke are just the reverse of the

slide valve events. This is, as previously explained, because the slide valve has outside admission, and inside exhaust, and may be readily understood from the following.

When the piston valve moves in the direction of the arrow A, Fig. 54, it will uncover the left-hand port,

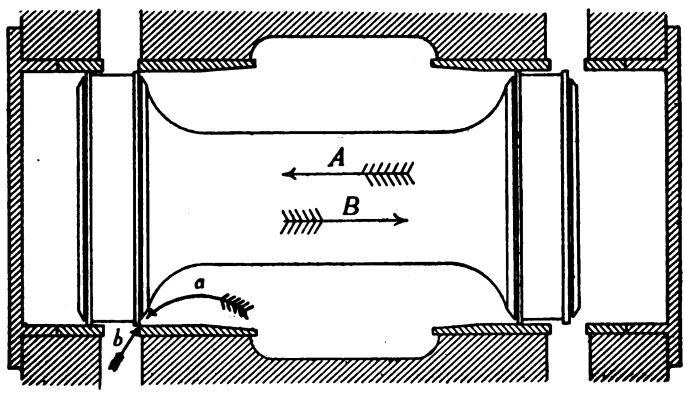


FIG. 54.

while a similar movement of a slide valve would mark the commencement of exhaust, but, with the piston valve steam is admitted from the inside of the valve at the point indicated by the arrow *a*. Admission begins here and will continue until the valve reaches the end of its travel to the left, and returns, in the direction of the arrow B, to cut-off—the position it occupies in the illustration—where the admission of steam to the cylinder

ceases, and cut-off takes place at the point *b*. With the slide valve compression would now begin.

Now directing our attention to Fig. 55, we will assume the valve continues its travel in the direction of the arrow until the outer edge of the valve coincides with the outer edge of the steam port indicated by the arrow *a*. It

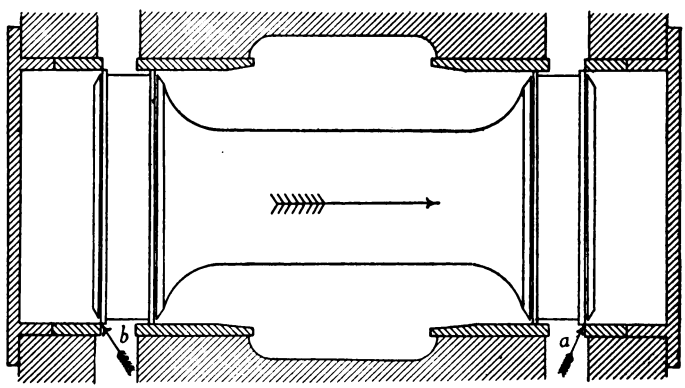


FIG. 55.

must be apparent that the steam which has not been exhausted from the right-hand port will be trapped in the cylinder, and must be compressed by the advancing piston which is also moving in the direction of the large arrow. This point, compression, corresponds to the point of cut-off with the slide valve.

It will be observed that the valve is now in its mid-position, for the two outer ends of the valve register with the outer edges of the steam ports; therefore, as

the valve continues its movement to the right the exhaust of steam will commence, in the left-hand port at *b*, whereas the movement of a slide valve in the same direction would allow the admission of steam in the left-hand port.

From this brief explanation of the difference in operation between the movements of a piston and a slide valve, the effects of over-travel, lap, lead, etc., with the piston valve, must be apparent.

Piston Valve Cylinders.

Piston valves can be located in almost any position with regard to the cylinders, and may be made any length desired. As a result the steam ports of piston valve cylinders should be made short and direct, and the steam chest placed close to the cylinder bore, leaving only sufficient room for the barrel flange to be turned for the head casting. Piston valve cylinders weigh less than first class slide-valve cylinders, and are more suitable to curved lines than the type used with slide-valves.

The Allfree Piston Valve Cylinders.

These cylinders embody firmly established principles which are applicable to the locomotive, and vary but little from ordinary practice, common standards being maintained. It may be noted, however, by referring to Fig.

56, that this type of cylinder is only arranged for inside admission valves.

The only essential difference in their general construction and that of other common designs is the use of the compression valve, and, as this positively actuated valve

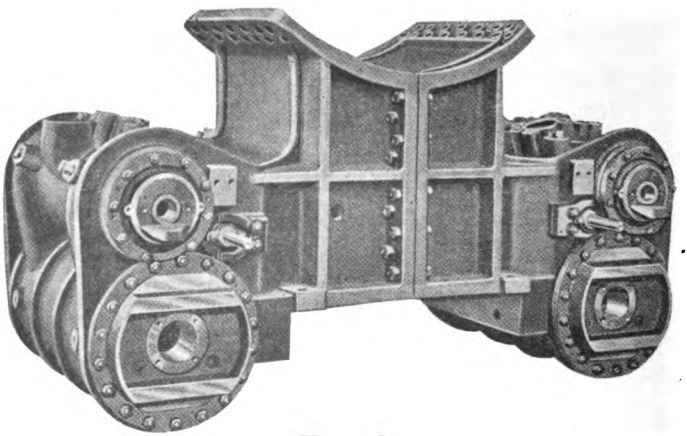


FIG. 56.

eliminates the necessity for, and takes the place of, by-pass and relief valves, there is no complication or multiplication of parts.

The Allfree Piston Valve.

This valve requires a special design of cylinder and steam chest (shown by Fig. 56), so arranged that there is a cylindrical opening between the main valve and the piston cylinder, in which an auxiliary compression valve

operates. The main valve is of the ordinary piston type, communicating with the cylinder through short, straight ports. The compression valve also is of the piston type,

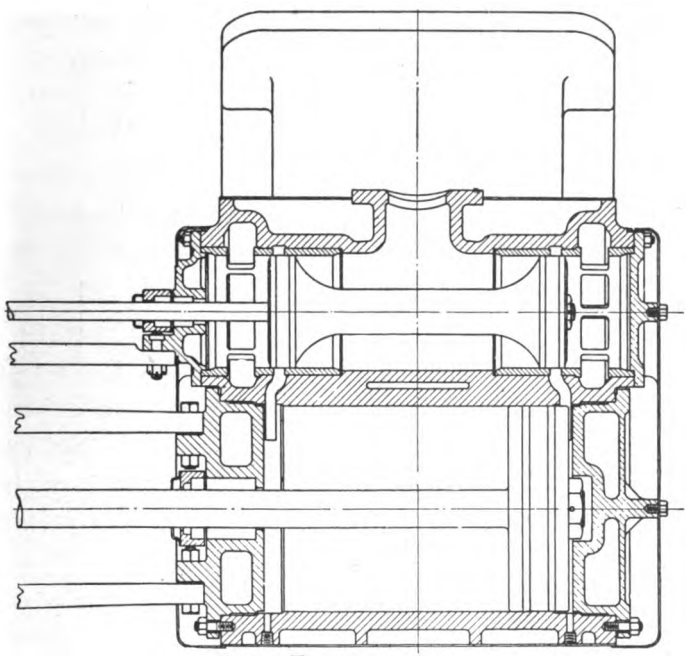


FIG. 57.

smaller than the main valve, and is in communication with the cylinder. Fig. 57 shows a section through the main valve and cylinder barrel.

The main valve operates just as the common inside admission piston valve, opening and closing the ports to

the admission of steam, except that it is given about 3/8 inch exhaust lap. This delays release, and therefore increases expansion. With the ordinary valve this would not be practical, for if exhaust were delayed too long, the steam would not all escape from the cylinder, and excessive back pressure would result. With the Allfree arrangement, however, the auxiliary valve, although it has no effect on admission or cut-off, opens to exhaust at the point of release of the main valve, and with this double opening the used steam is easily and rapidly released from the cylinder.

Delayed release, however, accounts for only a part of the increased expansion. This valve admits less steam for a given cut-off than the ordinary valve, because there is only about one-fourth the usual amount of clearance between the piston and the cylinder head. It is easily seen that this reduction of clearance increases the period of expansion.

With the common valve, when exhaust ends compression commences, and, with one-third cut-off position, continues for nearly one-third the entire stroke of the piston. The compression valve, however, in the Allfree arrangement, remains open after the main valve closes to exhaust, and the used steam continues to escape. A section through the compression valve is shown in Fig. 58.

The compression valve remains open until the piston has completed all but about two inches of its stroke, when the valve closes and compression commences. This

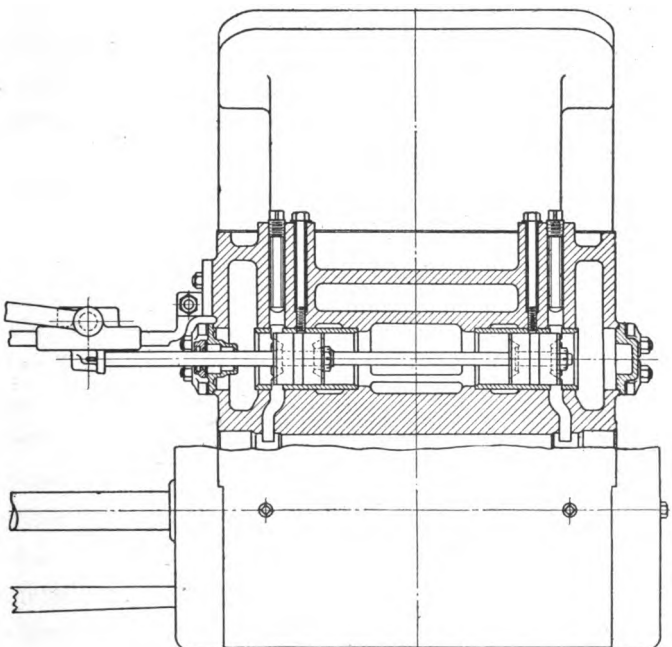


FIG. 58.

greatly reduced compression adds to the unobstructed stroke of the piston, reducing negative work in the cylinder. It does not, however, produce a shock on the piston and its connections, as might be imagined, for the reduced amount of steam is compressed in a correspond-

ingly smaller space, due to reduced clearance, and this has the same cushioning effect as the greater amount of steam compressed in the larger clearance space of the ordinary cylinder.

It is also claimed that a saving in steam results from the smaller clearance space having to be filled by admission steam, when the pressure of the compressed steam does not equal the initial, or boiler, pressure.

This valve was patented July 9, 1912, by Mr. Edwin H. Allfree, of Chicago, Ill.

American Semi-plug Piston Valve.

This is a special form of piston valve, patented November 17, 1908, by Mr. John T. Wilson, of Jersey Shore, Pa., which was designed to embody the advantages and to eliminate the disadvantages of the solid plug type of valve, which becomes leaky with wear, and the snap ring type, which tends to wear the valve cage unevenly.

The aim of the designer was to produce a valve which would automatically obtain a correct contact pressure between the packing rings and bushing, and then lock the rings in place, while steam chest pressure continued, by means of wedge rings under steam pressure.

A detail section of this valve is presented in Fig. 59 and the construction and operation of the valve may be described as follows:

The two snap rings numbered "1," are put in under tension, and are, therefore, self-adjusting; their outside walls are straight, and fit against the straight walls of the follower and spool. The inner walls of these snap rings are beveled, forming a cone. Next to the snap rings are wall rings, numbered "2," the outer faces of which are beveled to fit the cones of the snap rings.

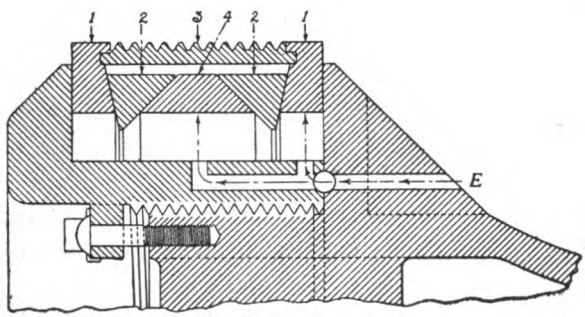


FIG. 59.

These are called "wall rings" because they form the inner walls for the snap rings. These wall rings are uncut, non-expansible steel rings. Between these wall rings, in the center, is placed a double coned expansible ring, called a "wedge ring," numbered "4," the sides of which are beveled to fit the inner sides of the wall rings. This wedge ring is put in under tension, and its tendency to expand crowds the two wall rings laterally against the cone sides of the snap rings, numbered "1"; this prevents lateral wear of all rings. A wide ring, num-

bered "3," interlocks with the snap rings, and completes the packing. The wide ring performs two important functions: First, it carries the snap rings across ports while the engine is drifting, and, second, it keeps the snap rings parallel and of equal diameter, so that the wide ring and the two snap rings form a flexible, yet wide, packing ring. The snap rings and the wide, or separating, ring expand and contract as one ring. All rings are free to turn around.

The principle of the valve is to control the frictional contact, or packing, against the cage by a system of wedges operated by the steam chest pressure, the wedges being in the form of cones, expansible and non-expansible. Now, having the principle of the valve fixed clearly in mind, we will note its operation.

The degree of angle on the cones, it will be observed, is much greater on the double tapered wedge ring than on the snap rings. These angles are so calculated that, while the pressure is underneath all the rings, the leverage of the double tapered wedge ring, crowding the solid wall rings against the cones of the snap rings, is just sufficient to prevent the snap rings from further expansion, but not sufficient to reduce the snap rings in diameter. By a little consideration of the effect of changing the degree of angle, it will be observed that the frictional contact of the snap rings against the valve

chamber depends entirely upon the angles, and it may, therefore, be regulated to any desired degree.

By observing the locking effect of the double tapered wedge ring, when it is expanded by pressure, thereby crowding the two solid wall rings laterally, and holding them, as it were, with a predetermined force against the cone sides of the snap rings, it will be readily realized that by sufficiently increasing the angle of the cones on the sides of the snap rings and making the solid wall rings to correspond, the force of the double tapered wedge ring, crowding the wall rings laterally on a sharper cone would be sufficient to close up, or decrease the diameter of the snap rings, regardless of the pressure under them; under these conditions the leverage would be too great, and would permit a blow over the outside of the snap ring. On the other hand, if the degree of angle on the snap ring were reduced, and the cone made flatter with wall rings to correspond, then the wedging power of the central wedge ring would not be sufficient to prevent the snap rings from being expanded by the pressure underneath them, and this would result in excessive friction against the valve chamber, caused by insufficient leverage to lock the rings. The same effect may be reached in either direction by changing the degree of angle of the double tapered wedge ring.

The action of the valve may be summarized as follows: When steam is admitted to the steam chest, it passes through the small holes, E, around the spool, and finds an outlet, first, under the snap ring, and, second, under the central wedge ring. From 14 to 18 holes are in each end of the valve. The two snap rings being interlocked through the wide ring, the velocity of the steam through these holes against the first snap ring insures the expansion of the packing to fit the valve chamber, and the velocity acting against the wedge ring places it in position for the pressure to lock up the rings. The force of velocity acts upon these rings before the accumulation of pressure locks them. While the engine is drifting, the packing rings contract, due to the lack of steam pressure, and the engine is thus relieved by allowing clearance space between the packing rings and the valve cage.

With this action clearly in the reader's mind, it will be seen that when the rings are locked in the cage they will remain of that diameter, unless they are locked in a large part of a worn cage; for instance, in which the movement of the valve forces the snap rings into the diameter of the cage at the smallest part. Under these conditions the snap rings would remain the smallest diameter of the cage in which the valve traveled back to the position of the original locking, and in that posi-

tion there would be a blow over the outside diameter of the snap ring. It will be seen, therefore, that this valve will never wear a cage out of true, and it will also be observed that it is very important that the valve be put into a true cage to begin with, and that the bushing must be kept true, to obtain satisfactory service.

The valve cage, or liner, is shown in Fig. 60, as is the extension valve stem and bearing. This extension

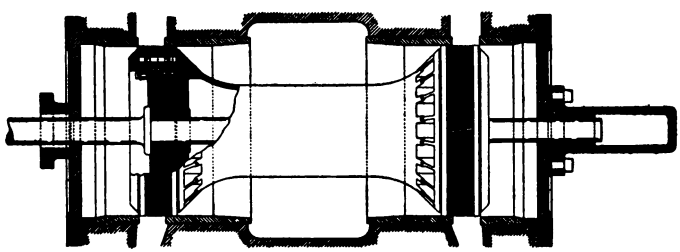


FIG. 60.

valve stem is essential to the use of this valve, as the valve must be carried central, or nearly so, and the spool must not be permitted to ride on the cage, or bushing, as this causes wear and a leaky valve. Although the arrangement is rather complicated, it has generally given satisfactory and efficient service in actual practice.

This valve is well balanced, and gives good service while new, but it is expensive to repair when worn. In some cases trouble was experienced with broken rings, due to the fact that the rings had worn to the breaking

point without renewal. The manufacturers now overcome this trouble by applying dowel pins to the connecting slot on the sides of the steam and exhaust rings, so that the valve will give warning by blowing before the rings have worn to the breaking point.

The Young Piston Valve.

This valve consists of a light spool-shaped cylinder, cast in one piece, the ends of which are of hollow construction, resembling somewhat the bushings applied to the ordinary piston valve chamber. These hollow ends are each made with two rows of openings, or ports, as shown in Fig. 61. The distance between the two rows of port holes in each end is so fixed that the valve alternately admits steam into the cylinder and allows it to escape, through the ports, in the same manner as the common inside admission piston valve.

The packing rings, instead of being placed in the valve to expand against the bushing, are fitted in suitable grooves in the valve chamber, on either side of each cylinder port, the outer surfaces of the valve ends being in sliding contact with them. The rings are of the split compression type, designed to contract against the cylindrical surface of the valve, and are spaced, by filling blocks, to their desired diameters.

This arrangement provides a valve which, after very slight wear to take up the original clearance between

the divided ends of the rings and their filling blocks, is practically steam tight. The operation of the valve is very similar to that of the ordinary solid plug form. However, due to its construction, the Young valve eliminates the many disadvantages of the plug type, more especially that of excessive weight. In fact, this valve is unusually light, for a twelve-inch valve will weigh but one hundred pounds, or about half the weight of

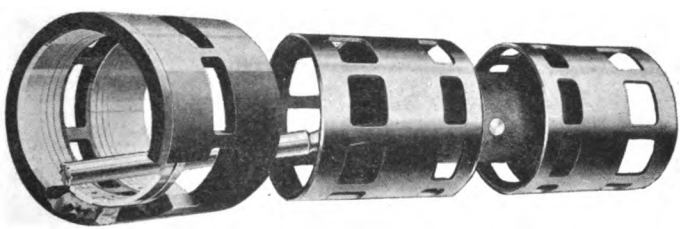


FIG. 61.

the common piston valve of the same diameter. This, of course, means a reduction in friction, and less stress upon the valve gear.

The valve is made with a special valve stem, which is provided with a knuckle. This knuckle is secured to the center of the valve spool by a fitted pin, and it is thus possible for the valve to align itself regardless of the alignment of the stem, eliminating uneven wear on the valve and the packing rings. This valve is shown in its position in the cylinder in Fig. 112.

This valve was invented by Mr. Otis W. Young, of Chicago, Ill., and is manufactured by the Pyle-National Company, of Chicago, Ill.

Piston versus Balanced Slide Valve.

As previously stated, the balanced slide valve was employed in locomotive service for several years before the piston valve was adopted, and the latter type was, by many, looked upon with disfavor. As a result each type of valve has its champions and there seems to be no unanimity of opinion regarding the merits or demerits of either type of valve.

The writer is not prompted by personal interest in advocating the advantages or disadvantages of either valve, so we therefore present an impartial review of what may be considered the weight of opinion regarding each type of valve at the present time.

Advantages of the Piston Valve.

The principal advantage of the piston valve over the balanced slide valve is the fact that it is more fully balanced, and therefore more easily manipulated from the cab. As a matter of fact, piston valves are generally handled with greater ease when the throttle is open than when it is closed, which is just the reverse of slide valves. The larger piston valves are almost as easily manipulated as the smaller ones.

The piston valve is probably more economical in steam consumption than the slide valve, but much depends upon the design and maintenance of the valve in question. If piston valves were given as much attention in the roundhouse as slide valves, they would, no doubt, show less steam loss from valve leakage than the balanced slide valve type.

Because they can be placed in almost any position with regard to the cylinder, the piston valves give flexibility of design. These valves can be placed above the cylinder, between the frame rails, or in nearly any position, thus they are adaptable to any form of valve gear. Piston valves can be made of any desirable length; therefore the steam ports can be made very short and direct.

With inside admission valves (piston valves are nearly always of this type; slide valves are not) the steam chest heads and packing are relieved of all pressure except that of exhaust steam, which puts very little stress upon these parts.

The cylinder casting used in connection with the piston valve is lighter and cheaper than that required for the slide valve, and the wearing surface, or bushing, is separate from the casting and can be renewed at a slight cost. The piston valve excels the slide valve in regard to maintenance cost.

The better balance of the piston valves, together with the reduced frictional resistance, relieve much wear and tear on the valve gear operating it, especially when superheat is used.

Disadvantages of the Piston Valve.

The piston valve has one decided disadvantage; it cannot relieve excessive pressure, or water in the cylinder. Many forms of relief valves have been used to accomplish this result, but they are expensive to maintain, and none give satisfactory results.

In some cases it has been found difficult to lubricate piston valves, more especially while drifting. Other defects are cut and broken packing rings, edge of spool breaking, liability to blow, excessive wear on bushing at short stroke, and friction due to the pressure of steam under the packing rings. The broken packing ring is not a serious trouble unless it catches in the port; the result is then a broken or sprung valve gear. Some roads report that, with piston valves, the main driving boxes wear so rapidly as to create side play and pound.

Advantages of the Slide Valve.

The chief advantage of the slide valve lies in the fact that it can relieve excessive pressure in the cylinder and steam chest by temporarily lifting from its seat. This

eliminates the troublesome relief valves used for this purpose with piston valves.

The balanced slide valve, which is practically the only kind now in use, tends to eliminate the excessive friction of the ordinary D slide valve. This is accomplished, as previously described, by the use of balance strips, and pressure plates, above the valve. This is really not an advantage of the slide over the piston valve, but rather an advantage which the balanced valve has over the plain D slide valve.

A few roads employ special methods of lubrication, and obtain satisfactory service from the use of balanced slide valves with superheat, but it it admitted that this service is excelled by the use of the piston valve.

Disadvantages of the Slide Valve.

When moving on its seat, even a well balanced slide valve creates excessive friction. This is especially true with the use of large cylinder dimensions and high steam pressures common to modern locomotives. This friction causes rapid wear on the valve, valve seat and gear, and it is very uncommon for an engine with slide valves to run 25,000 miles before the valves require facing. The large steam chests necessary also cause condensation of steam, which accounts for the high water rate of engines so equipped.

There is sometimes an upward pressure on the face of the slide valve, which causes a rocking motion, thus wearing the corners of the valve. When these corners are rounded, the valve is not tight, but will leak steam.

Ordinarily, slide valves cannot be operated successfully with superheat because of the high steam pressure. Thus, with the advent of the large and more modern locomotives, the balanced slide valve is fast giving way to the more efficient piston valve.

The cost of maintenance of the slide valve exceeds that of the piston type. The balance feature must be retained in good condition, and this necessitates much care. False valve seats are often necessary, when the valve seat wears to the limit. These are expensive to machine and apply, and are not altogether satisfactory.

Because the slide valve chest is subject to boiler pressure, it is no light task to prevent steam leaks in the joints at the top and bottom of the steam chest. The valve stem must be packed against boiler pressure, whereas the stem of the piston valve is subject merely to exhaust pressure. If steam leaks, lubrication is to a certain extent destroyed.

The pressure plate of the slide valve is often cut by the end balance strips. Many roads weaken the springs under these strips, or reduce their number, to overcome this difficulty.

The slide valve requires much more space than the piston valve, and adds greatly to the weight of the cylinder casting.

The most common failures experienced with the use of the slide valve are broken valve yokes and stems, due usually to cut valve seats, faulty lubrication, or defects in the balance feature of the valve.

Universal Valve Chest.

An ingenious arrangement for the application of piston valves to cylinders primarily designed for slide valves, was invented July 14, 1914, by Mr. John E. Muhlfeld, of Scarsdale, N. Y., and Mr. Hal R. Stafford, of Plainfield, N. J.

We understand that the device has been successfully adapted to every design of slide valve cylinder in service. In fact, it can be designed for every combination, for the application of the device permits the use of superheated steam with the old cylinder, at almost the same cost as cylinders originally designed for piston valves, with the same effect and power, for either inside or outside admission valves, and may be used with any type of valve gear in use at present.

There are two distinct types of this valve chest, viz.:

In type A, Fig. 62, a separate inner valve chamber is jointed to the valve seat, independent of the outer chest,

which is seated on the cylinder apron in the usual way. Thus the regular inlet ports to the steam chest may be utilized. This type is also designed for use with either inside or outside steam pipes.

On the other hand, type B, Fig. 63, is designed for use where new outside steam pipes are to be applied.

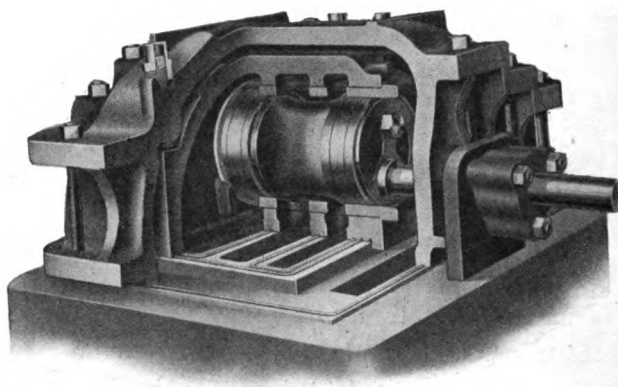


FIG. 62.

This type is made in one piece, is joined to the valve seat only, and by a system of joint wires set in grooves, around the chest and each separate port, is rendered steam tight.

An end view, in cross-section, of both valves is shown in Fig. 64. On the left is shown the type B, Fig. 63, which calls for the application of new outside steam pipes. On the right, type A, Fig. 62, designed for use

with either inside or outside steam pipes, is represented. This combination of the two figures clearly illustrates the difference in construction of the valve chest to be used with either type.

A special feature claimed for the invention is that it

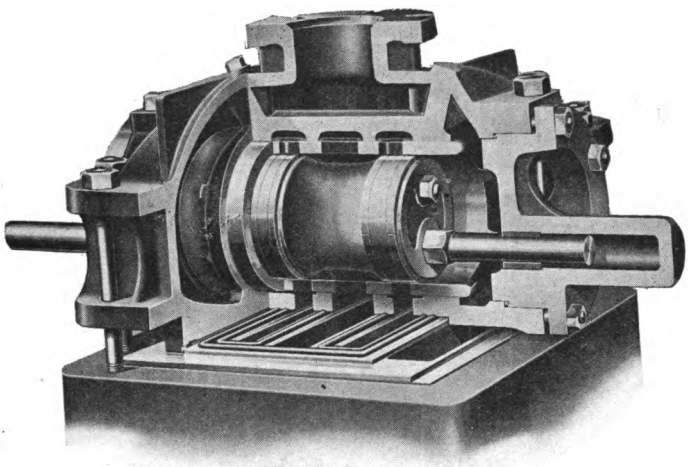
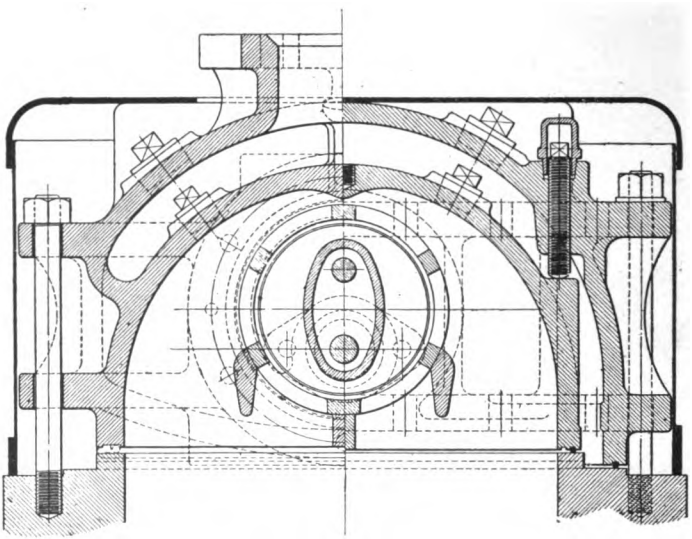


FIG. 63.

permits the use of a valve of minimum diameter, because the steam ports are of such form that the entire circumference of the valve is effective.

The advantages gained by the use of the small valves will be readily appreciated by practical mechanics, for the decrease in size reduces friction and stresses proportionally, and results in a better steam distribution.

The Economy Device Corporation, of New York City, are the manufacturers of this device.



TYPE B TYPE A

FIG. 64.

A Straightway Piston Valve Arrangement.

A new method, or arrangement, of restricting the size of the valve for ordinary piston valve cylinders, in accordance with the established principles of the flow of gases, without sacrificing the amount of work generally secured by the use of a larger valve, is described and illustrated herewith.

With the use of this device it is unnecessary to make the circumference of a piston any greater than the length of a slide valve port, for the beneficial effect is obtained by properly directing the flow of steam to and from the valve so that every square inch of the bushing

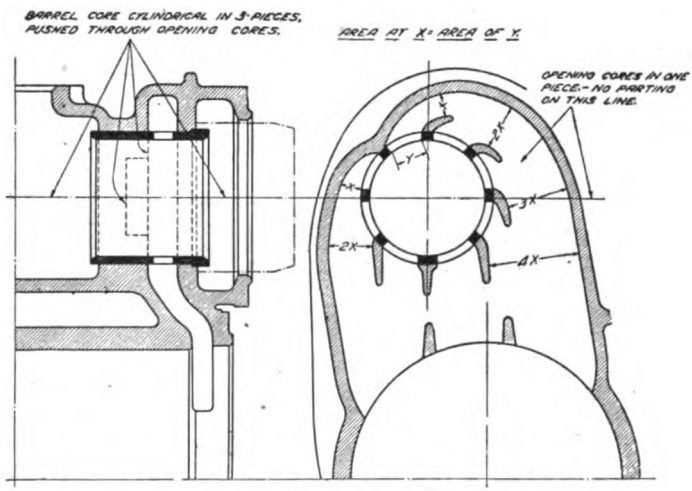


FIG. 65.

port will become effective. This permits the use of a valve only half the size commonly used, and, at the same time, facilitates the flow of steam to and from the cylinder to such an extent that the engine is much smarter and faster.

Fig. 65 shows the application of these principles to new power, where the design of the cylinders is under

control. The letter Y equals the area of each bushing port, while X equals Y in area. Thus it will be seen that the steam port around the bushing is so arranged that, during the admission period, as steam issues from the valve, it is divided into a number of streams, de-

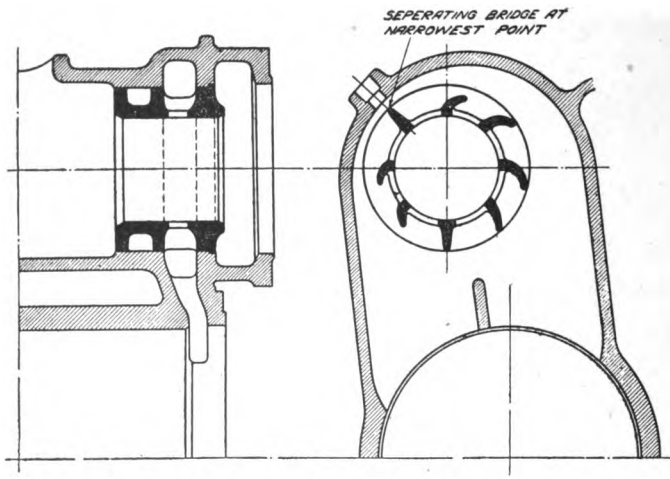


FIG. 66.

pending upon the necessary number of bridges in the bushing. These streams are directed by the ribs so that they do not interfere with each other, but join into one smooth flowing whole when the main passage is reached. During the exhaust stroke the steam, as it flows from the cylinder to the valve, is divided by the ribs into a number of equal streams, one from each port in the bushing.

This absence of all quick turns, or other obstructions, speeds up the flow of steam, and, with an eight or ten inch valve, gives a higher initial pressure and a better admission line than can be obtained with the ordinary design of port with a fourteen or sixteen inch valve.

Fig. 66 shows the method of applying small valves to existing engines on which valves of much larger diameter were originally used. In this case a new bushing of special design, in which the directing ribs are cast as a part of the bushing bridges, is applied in place of the original bushing. This arrangement makes it possible to secure the advantages of the smaller and lighter valve, and gives a steam distribution at least equal, and in most cases superior, to that of the larger valve.

This device is also manufactured by the Economy Devices Corporation, of New York City, N. Y.

Sheedy's Outside-steam-pipe for Piston-valves.

This device was invented by Mr. Patrick Sheedy, of Los Angeles, Cal., and a patent was issued to him on September 5, 1916.

The object of this invention is to effect a material and substantial economy in the modernization of existing locomotives which relates to improvements in cylinder construction, by a simple and inexpensive method of converting existing cylinders in which steam is supplied through an inside steam pipe, to a steam passage in the

cylinder saddle, into cylinders, adapted to be supplied with steam through an outside steam pipe.

The advantages of effecting steam supply to locomotive cylinders through what are known as outside steam

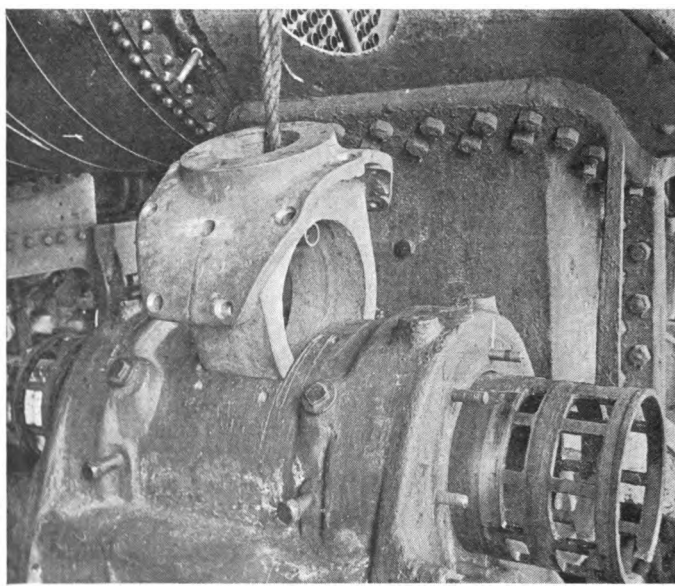


FIG. 67.

pipes, particularly in facilitating draft by relieving the lower portion of the smoke box from obstruction, eliminating steam leaks through the bottom steam pipe joints in the smoke box, and promoting the application of superheaters, are so well and thoroughly recognized, that

in practically all new locomotives the cylinders are provided with outside steam pipes.

When it has been considered necessary to modernize existing locomotives, by the application of recent improvements in structural detail, the inside steam pipe cylinders have been scrapped and new cylinders with outside steam pipes substituted, but such a change can only be made at considerable cost.

This invention provides a method of applying outside steam pipe cylinders of the latest pattern to old style locomotives with inside steam pipe connections, without a change of the original cylinder at considerable saving.

Fig. 67 shows a section of a cylinder, commonly used in locomotive construction, and Fig. 68 another view of the same cylinder equipped with the device under consideration. First a rectangular transverse passageway is cut in the top of the valve chest, symmetrically with the middle transverse plane thereof, and an integral bushing, termed a finger ring bushing, is then dropped through this opening into the valve chest, and secured therein with its horizontal bore in line axially with the valve bushings. The bushing is open at the bottom on its opposite sides to the valve bushings, and is open at top which is adapted for connection to the lower end of an outside steam pipe, a ball joint ring being interposed between it and the steam pipe. The bushing is

connected to the valve chest by bolts passing through a circumferential flange on the bushing the under side of which is curved in conformity with the adjoining outer surface of the valve chest. The bushing is then

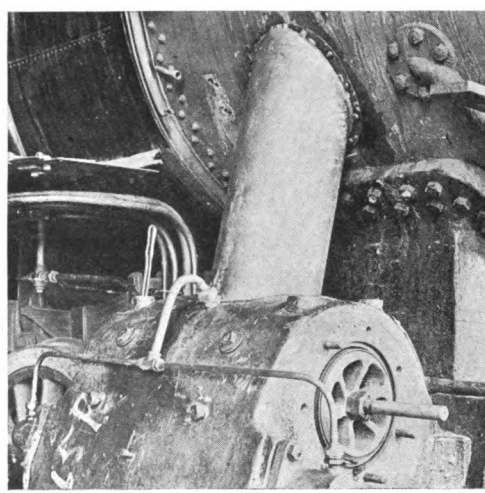


FIG. 68.

bored out in line axially with valve bushings, and valve bushings are pressed steam tight into opposite ends of the valve chamber in which they are permanently located.

The original steam passage may then be filled with a mixture of cement and iron borings to support the bushing and prevent deterioration due to the entry of moisture into the unused passage.

It will be seen by the above method of reconstruction, which can be effected at comparatively slight expense, an outside steam pipe may be applied to a locomotive cylinder, which was originally designed for inside steam pipe supply with the same facility as if the latter had been as constructed, adapted for such application, and in the improvement or modernization of the locomotive, in this particular, the very substantial economy of avoiding the abandonment and scrapping of a very valuable pair of cylinders.

VALVE GEARS.

Introduction.

The necessity of a device for reversing, or changing, the motion of a locomotive was promptly recognized in the early days of locomotive construction, and numerous devices, which are known as "valve gears," have been invented to accomplish the result desired.

The first, or original, device employed to reverse a locomotive consisted of a single eccentric which was loose on the shaft, and could be turned by a hand wheel. When the steam was shut off and the engine brought to a stop, the engineer could go forward and loosen a nut, turn the eccentric around to the position desired, screw the nut tight to clamp the eccentric in its new position, and proceed in the opposite direction.

We assume the original device did not give entire satisfaction, however, and it may not have been popular with the engineers of that day, for another reversing valve gear, called the "gab-hooks," Fig. 69, was devised which employed two eccentrics and enabled the engineer to change the motion of the engine without leaving the cab. Each eccentric-rod had a notched hook attached to its forward end, and the back end of the valve-rod carried a pin on which the notched hooks

could be brought into contact to actuate the valve. When the upper hook was lowered to connect with the valve-rod pin the forward eccentric operated the valve, and vice versa.

The result of lowering or raising a hook to catch a pin in the end of the valve-rod to reverse the move-

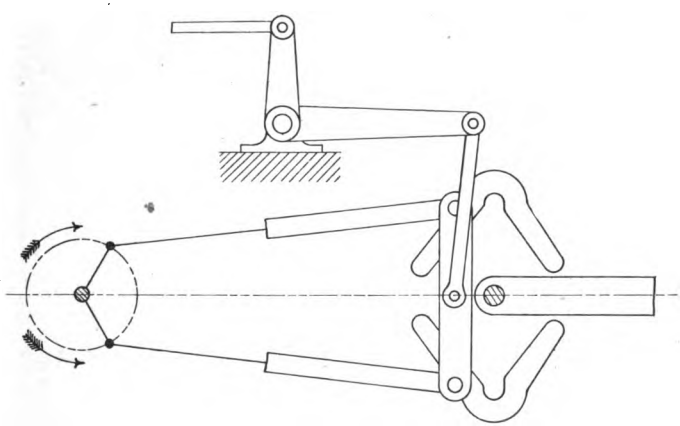


FIG. 69.

ment of a high speed locomotive of the present day can be better imagined than described.

The only noteworthy thing about this form of valve gear is that it was the first, and the last, to use hooks, and it introduced the principle, which has been adopted by numerous inventors since, of using two eccentrics.

The next step toward the improvement of valve gears was the introduction of the curved or straight link which

is at present employed in connection with practically all reversing valve gears. This was, in fact, a great improvement, not only in providing for the reversal of the locomotive, but because it provided means of allowing a variation in the arrangement of the valve mechanism, thereby permitting a more economical steam distribution in the cylinder.

Numerous other valve gears appeared from time to time, but they did not possess sufficient merit to supplant the Stephenson link-motion which was, for many years, used exclusively in American locomotive construction.

With the introduction of the powerful locomotives of the present day, however, it was found the Stephenson link-motion could no longer be used to advantage and, for structural reasons, it was replaced by the various types of outside valve gear, which will be described later on. We do not believe, therefore, that a description of it would be of advantage to the reader, so we shall confine our discussion to the modern valve gears in use today.

We deem it proper to first describe the pioneer radial valve gear which is most extensively used, and whose popularity has steadily increased since its adoption in American locomotive construction—the Walschaert.

THE WALSCHAERT VALVE GEAR.

HISTORY.

Before proceeding into a discussion of this well known form of valve gear, it seems to the writer that a brief history of the life and work of the inventor should be of interest to the reader, and with this idea in mind we present the following facts:

The inventor of this valve gear was born in Malines, Belgium, on January 21, 1820, and was named Egide Walschaerts. Some years later, for reasons unknown to the writer, he saw fit to omit the final letter *s* from his surname, so we shall do likewise in referring to him in this work.

About the year 1835, when the line from Brussels to Malines was opened, his native city was made the central point of the Belgium State Railways. The event proved to be the turning point in the career of young Walschaert, for the construction of the locomotive made a strong impression on his mind, which lasted until the end of his life.

In 1838, while a student in the College of Malines, he exhibited some models of a stationary and a locomotive engine of his own construction, at a local exposition. He continued his experiments with more or less success and secured a position as a mechanic in the

Belgium State Railway shops in Malines in 1842, and became a foreman in the shops at Brussels two years later, when his ability as an engineer had clearly demonstrated his qualification to rule the motive department, yet he was never permitted to rise to a higher official position.

In those early days the locomotives used in Belgium were imported from England, and they had only been in service a few years when Walschaert was made a foreman. But it is now evident that they did not meet with his approval, for he soon began work upon a new system of valve motion, which invention he completed in the year of 1844.

Under the rules of the Belgium State Railways, in force at that time, a mechanic, or foreman, was not allowed to patent a discovery or device for his own benefit and profit, because the product of his brain, as well as his hands, was considered the property of his employer. So to circumvent the rule Walschaert induced an engineer of the Belgium State Railways to seek a patent for him. The application for this Belgium patent was filed by his friend, Mr. Fisher, on October 5, 1844, and the patent was issued November 30, 1844, for a term of 15 years.

It may be stated here, to the credit of Mr. Fisher, that he never claimed credit, personal or otherwise, for the

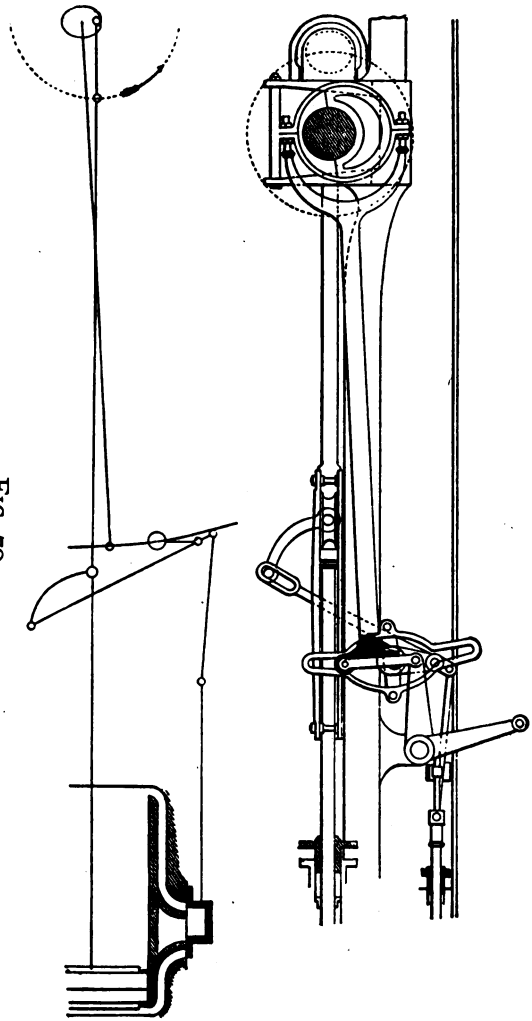


FIG. 70.

invention of the valve gear on which he secured a patent.

The rule referred to, however, did not apply to foreign countries, and Walschaert secured a patent on his valve gear in France on October 25, 1844, in his own name.

The invention immediately attracted attention in Belgium, but some time elapsed before it was adopted by the leading railways of Europe. Considerable effort was made to introduce the valve gear in America about 10 to 15 years after it was invented, but the construction of our locomotives and the existing conditions were not then favorable. About 1897 a few American locomotives were equipped with the gear and the results obtained during the following years were so satisfactory that the valve gear gradually worked its way into general favor.

The original design of Walschaert's valve gear, upon which a Belgium patent was issued to his friend Fisher, is shown in Fig. 70.

While there is considerable difference in construction of the mechanism patented in 1844, it is, in principle, similar to the valve gear in use today, which was constructed by Walschaert in 1848, and first used on locomotive No. 98 of the Belgium State Railways, when it ran from Malines to Brussels on September 2, 1848.

While it is true the link and combination lever are now usually placed in a different position to shorten the eccentric and valve stem, Fig. 71 of the original design shows, in a general way, the valve motion in use today, for quite a number of locomotives at the present time require an arrangement similar to that of the illustration.

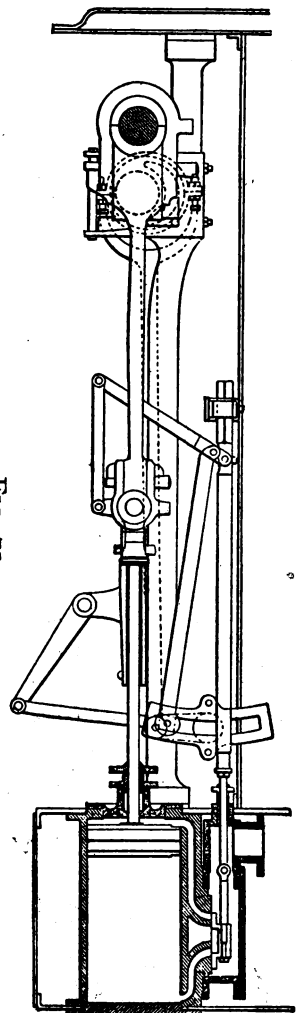


FIG. 71.

At the time Walschaert began his work, Sharp's valve-motion, which consisted of two eccentrics with forked rods, was the only system in extensive use, and it is very doubtful whether Walschaert ever saw the link motion credited to Stephenson, which was in fact invented by Howe in 1843, before he completed his own system.

The inventor's activity during the following years was confined, to a great extent, to the duties imposed on him as foreman, but

a number of other improvements and inventions conceived in his ingenious brain were used by practical men of that time.

It is now difficult to understand how Walschaert's inventive spirit was maintained in the face of the obstacles encountered, and the field to which his services were restricted. It certainly was not stimulated by any monetary reward, for he never received much, if anything, except a few medals, for the invention of his valve gear.

Then death deprived him of the pleasure of seeing the general adoption of his valve gear by the locomotives of the twentieth century, for he died on February 18, 1901, at Saint Gilles, near Brussels, in Belgium, at the age of 81 years and 29 days.

It has been well said that his work now meets requirements which did not exist during his lifetime and, for this reason, he is entitled to credit for solving a problem of, to him, a future generation.

Analysis.

The principal advantage of this gear lies in the accessibility of its parts, which are placed entirely outside of the driving wheels; this facilitates oiling, inspecting and cleaning operations which are frequently difficult in performing on locomotives equipped with the Stephenson link motion. Furthermore, in heavy en-

gines equipped with the Stephenson gear, the eccentrics must be made of large diameter to secure the required throw. This increases the velocity of the rubbing surfaces and tendency to heat, especially in the case of locomotives which have comparatively small wheels and are employed in high speed service.

In the Walschaert gear the various parts are pin-connected and are easily lubricated, hence troubles due to heating are reduced to a minimum. Furthermore, the Walschaert gear, as usually constructed, transmits the moving force of the valve in a practically straight line; consequently there is less springing and yielding of the parts than in the Stephenson link motion.

By removing the valve gear from between the frames, as in the case of the Walschaert valve gear, a better opportunity is afforded to introduce stronger frame bracing, and this reduces the possibility of the frame breaking. It is purposed here to explain the theory and action of the Walschaert valve gear, in as simple and plain a manner as possible, with the help of numerous illustrations.

In commencing the study, or entering into an analysis, of any particular form of locomotive valve gear, it must be assumed that the principle of the plain steam engine in its most primitive form is already understood, at least, of the valve itself. As to the valve, it is a study in

itself, and we must become acquainted with it before starting into the subject of the mechanism that operates it.

So far as the distribution of steam in the cylinder is concerned, the constant lead, which is a feature of this motion, is not considered objectionable, and it has some distinct advantages. Under such conditions it is possible to determine the amount of lead the engine should have at the most economical point of cut-off. This point once determined, and so designed, it cannot be altered by anyone in the shop or roundhouse. Another advantage is that it prevents valve setters from attempting to produce results by moving the eccentrics into improper relations one to another.

The constant lead of the Walschaert valve motion prevents the sealing of the cylinders by the piston valve when the piston is at the end of its travel, or approaching it, whereas with the link motion, either by derangement or excessive wear, the valve laps the ports at the end of the stroke, thereby causing excessive compression and many other troubles.

Another feature of this motion, which appeals to the engineers, is the ease of handling the reverse lever when the locomotive is running at a high rate of speed.

Lead is given in order that steam may be admitted between the piston and the cylinder head, toward the

completion of the stroke, as a means of cushioning the piston and thus tempering the sudden reversion of its motion. There are many cases in which the variable lead is preferred; for instance, with a variable lead, the longest cut-off in starting can be obtained, combined with the proper amount of lead at the ordinary running cut-off.

In the case of passenger locomotives particularly, a steam distribution like this is often desirable. The Walschaert valve gear, as already stated, may be designed to give a variable lead. This practice has recently been followed in a number of instances. The favorable results for starting are, however, obtained at the expense of the distortion of the valve events in back motion by robbing one to favor the other.

For this reason the Walschaert valve gear, with variable lead, is suitable only for passenger and fast freight locomotives and not for slow freight or switching locomotives. With a variable lead, so arranged that the lead increases as the reverse lever is hooked up, the eccentric crank lags behind the correct position for a constant lead; in other words, it is so set that the link is not in its central position when the crank pin is on the center. The general arrangement of the Walschaert valve gear depends largely on the general design of the locomotive.

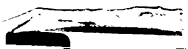
Construction.

The reliability of the Walschaert valve motion on locomotives, when properly applied, has practically put the former standard (the Stephenson motion) out of use. It is, therefore, considered that a complete elementary explanation of the theory of this gear, as applied in practice, is desirable.

This will involve considerable repetition of data which has been previously published, but the object is to show more intimately the fundamental co-relation between the various parts that make up this gear, in order to make as plain as possible its extreme simplicity and to enable even a layman to master the gear without further instruction.

Like any other device, the Walschaert valve motion is merely a development of some more simple form. A clear understanding of the underlying principles of this valve gear is, therefore, best obtained by starting with the original form and tracing the various steps in the development.

Fig. 72 represents a simple form of valve motion. This consists of a single crank eccentric, driving a plain valve without lap or lead, by means of an eccentric rod directly connected to the valve stem. Assume that the engine is to run forward; with the main pin on the back center, as shown, the eccentric crank pin must



be on the top quarter. The valve will then be in a central position on its seat with all ports closed. The engine could not, therefore, start of itself when the throttle was opened. Connect another engine of the same kind to the wheel on the other side of the axle. If, then, the main pins are set at right angles to each other, as in the case of the locomotive, the valve of the left hand engine would be in a position to admit steam behind its piston and start the engine forward. The eccentric crank of the engine shown in the illustration would then move its valve forward. This would uncover the back steam port and admit steam behind the piston.

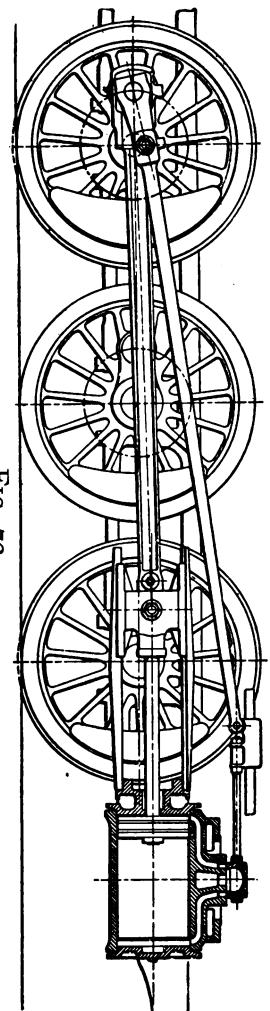


FIG. 72.

Such an engine, however, would run in only one direction. If, for example, the wheel was pinched backward to start the engine backward; with the eccentric crank in the position shown, the valve

would be moved back. Steam would be admitted ahead of instead of behind the piston.

In order to make the engine run backward, the eccentric would have to be on the lower quarter. In such a case, if the wheel was turned to the left or backward, the eccentric would be a quarter of a revolution ahead of the main pin. It would thus move the valve forward, opening up the back port and admitting steam behind the piston.

The introduction of some means for reversing is the first step necessary in the development of this simple form of engine. This can be accomplished by introducing between the eccentric crank and the valve stem a beam pivoted at its center as shown in Fig. 73. Here the link is the beam.

Referring to diagram "A," Fig. 73, with the valve stem connected to the lower end of the link and the main pin on the upper quarter, the valve is in the position to start the engine forward. Assume, on the other hand, that the main pin and eccentric crank remain in the same positions and the valve stem is connected to the upper end of the link, as shown in diagram "B." The valve is then in its extreme position to the left. When the throttle is opened, the engine will run backward. It is evident, then, that a radius rod connected to a block which slides in a curved slot in the link, as shown

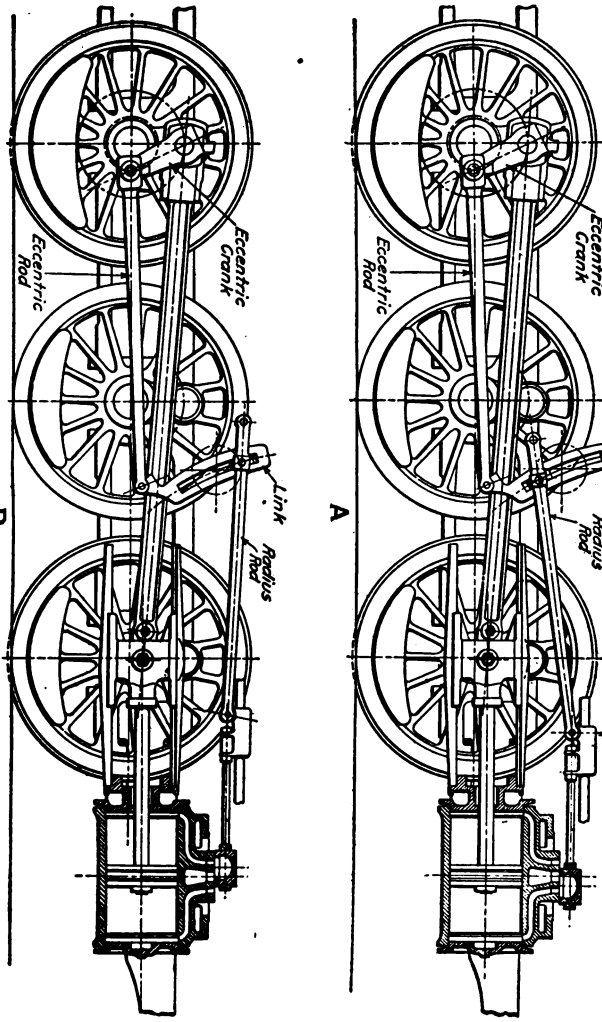

Eccentric Crank
Eccentric Rod
Link
Radius Rod
A
Eccentric Crank
Eccentric Rod
Link
Radius Rod
B

FIG. 73.

in the diagram, and the necessary mechanism for raising and lowering the block, would give an engine which could be reversed.

This engine, however, is far from an efficient machine. The valve does not close the admission port until the main pin is on the center, at which time the valve is in its central position on the seat. Steam will thus be admitted to the cylinders throughout the full stroke of the piston, irrespective of the travel of the valve.

In order to govern the period for the admission of steam; or, in other words, to change the cut-off; a different design of valve must be used. The valve must be given lap. Also, the valve should have lead, at least at the ordinary running cut-off.

If lap and lead are given to the valve, the valve motion shown in Fig. 73 must be so modified that, when the crank pin is on either of the centers, the valve will be advanced a distance equal to the lap plus the lead.

A study of Fig. 74 shows that this advance of the valve cannot be obtained by any change in the position of the eccentric crank relative to the main pin. With a single eccentic crank, as shown, the engine could not be reversed if the eccentric crank was not a quarter of a revolution from the main pin.

Fig. 74 shows the same valve motion as Fig. 73; except that the valve has one inch outside lap and the

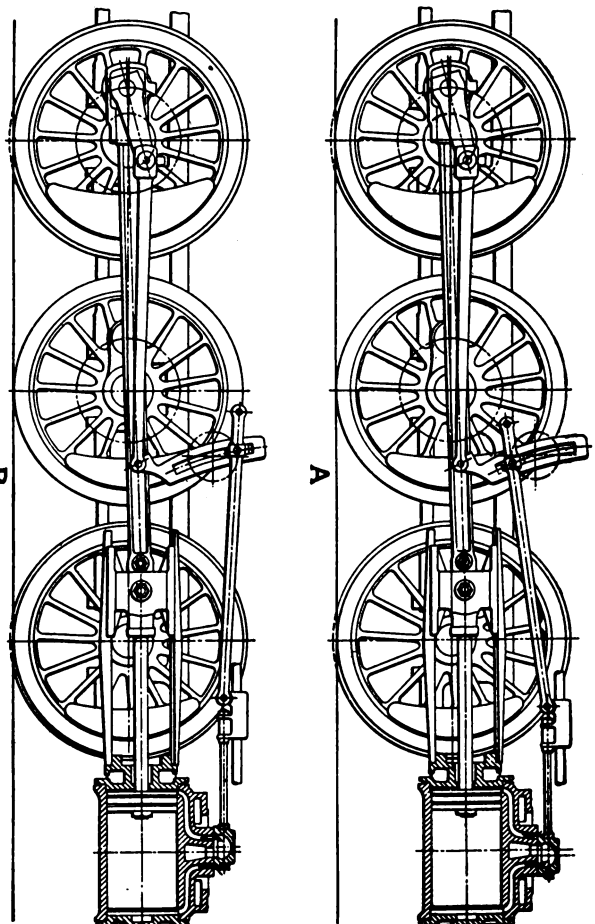


FIG. 74.

eccentric crank has been advanced to more than a quarter of a revolution ahead of the main pin. With the link block at the bottom of the link, as in diagram "A," the advance given to the eccentric will have moved the valve forward a distance equal to the lap plus the lead, and the engine will run forward. But if the link block were moved to the upper end of the link, as in diagram "B," the valve would be moved back. Steam would then be admitted to the front end of the cylinder when the piston is at the end of the back stroke. The engine would then stall and, therefore, could not be reversed.

In the Walschaert valve gear, the motion for providing lap and lead to the valve is derived from the main pin through the lap and lead lever, which is connected to the crosshead.

The introduction of the lap and lead lever is the next step in the development of the Walschaert valve gear.

Referring to Fig. 75, when the link block is in the center of the link, as in diagram "A," there will be no movement of the radius rod as the link is swung back and forth by the eccentric. Assume that the radius rod is connected with the lap and lead lever at the point "R." Also assume that the upper end of the lever is connected with the valve stem crosshead at "V," and

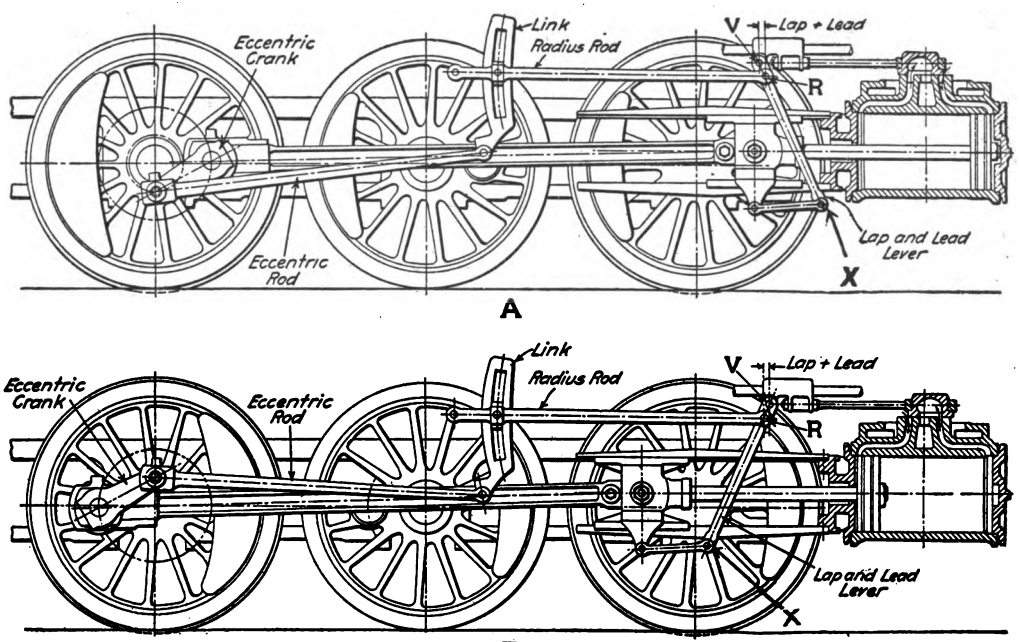

Eccentric Crank
Link
Radius Rod
V
Lap + Lead
R
Eccentric Rod
Lap and Lead Lever
X
A
Eccentric Crank
Eccentric Rod
Link
Radius Rod
V
Lap + Lead
R
Lap and Lead Lever
X
B

FIG. 75.

the lower end to the crosshead arm by a short link, as shown. With such a construction, as the crosshead moves back and forth, point "R" being stationary, the point "V" will rotate about it. The valve will thus be moved back and forth.

With the main pin on the forward center, as in diagram "A," the angle assumed by the lap and lead lever has moved the valve back a sufficient distance to uncover the front port. When the main pin is on the back center, as in diagram "B," the inclination of this lever has opened the back port.

Therefore, the motion of the valve is derived from two distinct sources, viz.: the eccentric crank and the main crank. The former gives the travel to the valve minus the lap and lead. This motion is transmitted through a slotted link oscillating on a fulcrum, in which slot the link block is moved at will above or below the fulcrum. By this means the motion of the engine can be reversed so as to run either forward or backward and it may, therefore, be called the reversing motion of the gear. The lap and lead motion, imparted to the valve by the main crank, is not reversible, as it is symmetrical with relation to the dead center position of the crank in whichever direction the engine is running, and is always constant, regardless of the position of the link block.

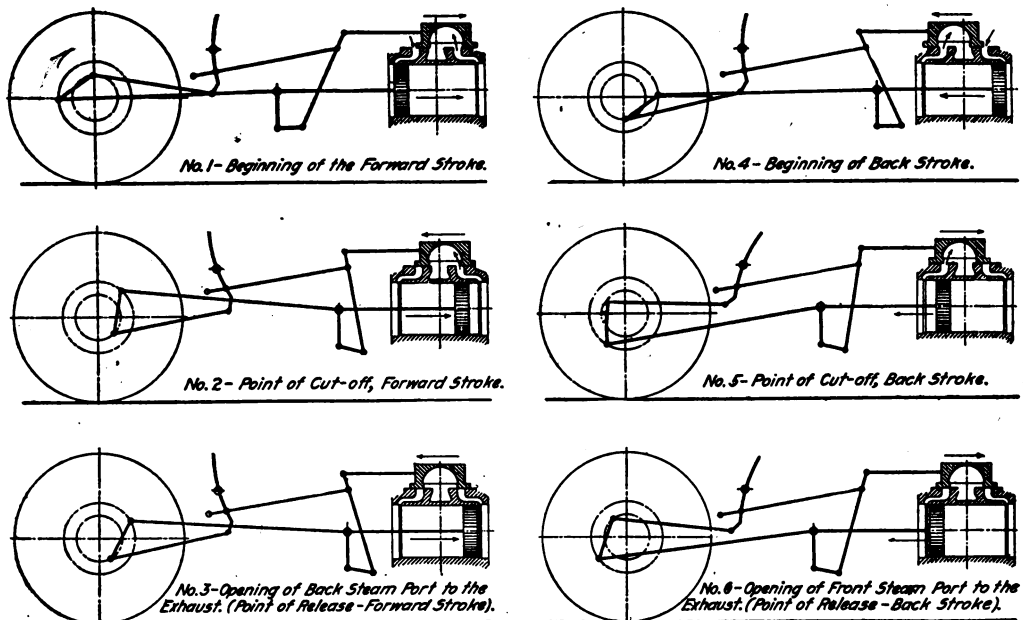

No. 1 - Beginning of the Forward Stroke.
No. 4 - Beginning of Back Stroke.
No. 2 - Point of Cut-off, Forward Stroke.
No. 5 - Point of Cut-off, Back Stroke.
No. 3 - Opening of Back Steam Port to the Exhaust. (Point of Release - Forward Stroke).
No. 6 - Opening of Front Steam Port to the Exhaust. (Point of Release - Back Stroke).

FIG. 76.

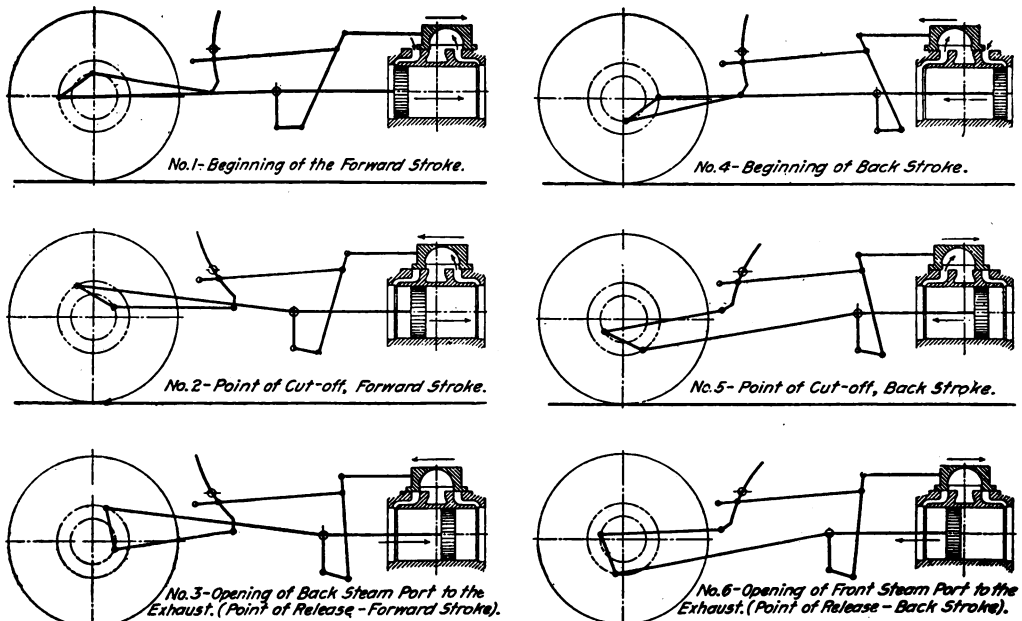

No. 1- Beginning of the Forward Stroke.
No. 4- Beginning of Back Stroke.
No. 2- Point of Cut-off, Forward Stroke.
No. 5- Point of Cut-off, Back Stroke.
No. 3- Opening of Back Steam Port to the Exhaust. (Point of Release - Forward Stroke).
No. 6- Opening of Front Steam Port to the Exhaust. (Point of Release - Back Stroke).

FIG. 77.

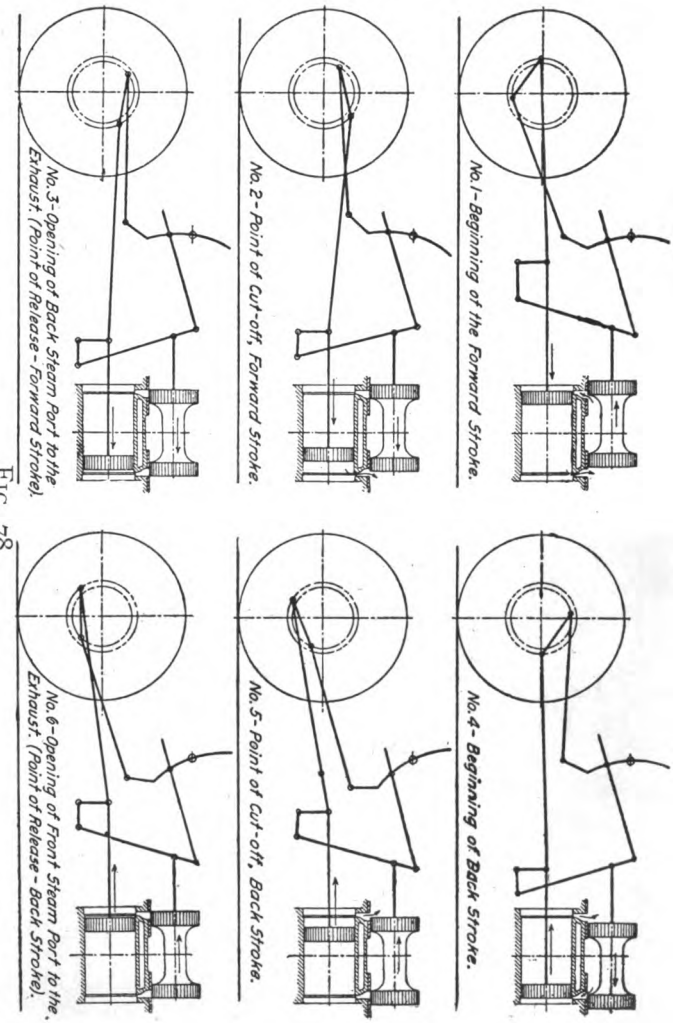

No. 1- Beginning of the Forward Stroke.
No. 2- Point of Cut-off, Forward Stroke.
No. 3- Opening of Back Steam Port to the Exhaust. (Point of Release- Forward Stroke).
No. 4- Beginning of Back Stroke.
No. 5- Point of Cut-off, Back Stroke.
No. 6- Opening of Front Steam Port to the Exhaust. (Point of Release- Back Stroke).

FIG. 78.

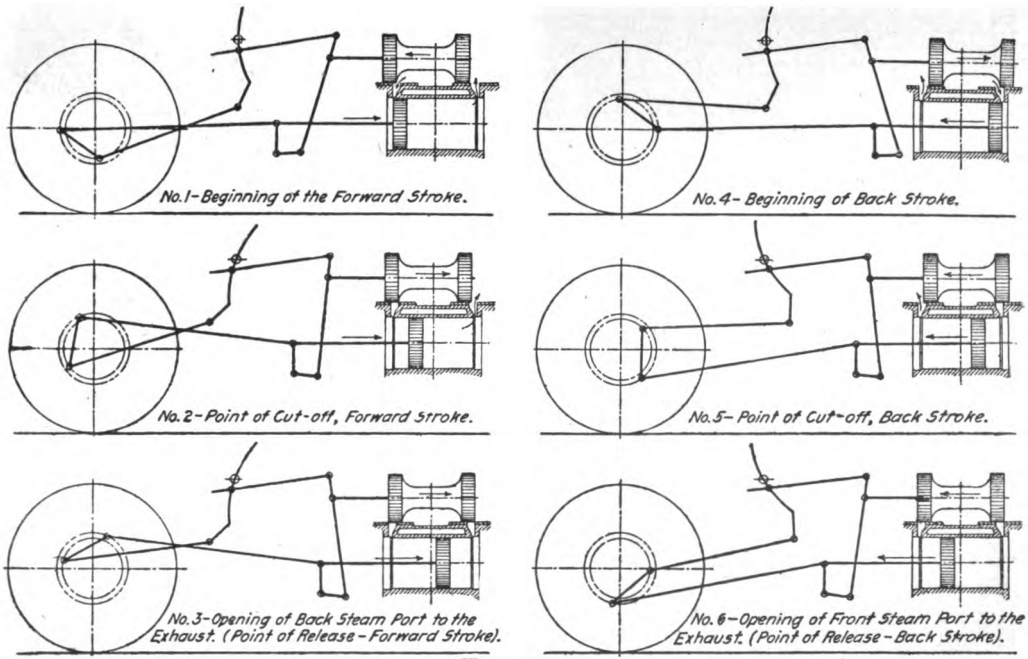

No. 1 - Beginning of the Forward Stroke.
No. 4 - Beginning of Back Stroke.
No. 2 - Point of Cut-off, Forward Stroke.
No. 5 - Point of Cut-off, Back Stroke.
No. 3 - Opening of Back Steam Port to the Exhaust. (Point of Release - Forward Stroke).
No. 6 - Opening of Front Steam Port to the Exhaust. (Point of Release - Back Stroke).

FIG. 79.

The action of the Walschaert valve gear as a whole may be best learned by tracing the movement of the valve through a complete revolution of the wheel.

Figs. 76 to 79 show a series of diagrams representing different positions of the crank pin. For the sake of simplicity, the valve and cylinder are shown in section. The other parts of the gear are represented by their center lines and center points only. These diagrams are purposely drawn out of proportion. The valve and the eccentric throw have been enlarged in order to show more clearly the positions of the edges of the valve relative to the edges of the cylinder ports.

In Fig. 76 the valve has outside admission; and the motion is represented with the reverse lever in full gear forward. Fig. 77 represents the same arrangement of valve motion. In this latter case, however, the reverse lever is hooked up and the engine is cutting off at about 25 per cent of the stroke.

Figs. 78 and 79 represent the Walschaert valve gear as arranged for piston valves having inside admission.

In the Walschaert valve gear, the valve, as previously explained, receives motion from two distinct sources: First, from the eccentric crank. This gives the valve its full travel. Second, from the lap and lead lever. This gives the valve its lead. The valve would receive a travel from this source equal to twice the lap

plus twice the lead, even if the eccentric rod was disconnected.

Considering diagram 1, Fig. 76, the valve has outside admission. Consequently, forward motion being taken from the bottom of the link, the eccentric crank leads the main pin. Also the radius rod is connected to the lap and lead lever below the valve stem.

The link is in its central position. Therefore, the valve would likewise be in its central position, if it were not for the motion given to it by the lap and lead lever. The travel of the crosshead to the back end of the stroke has caused this lever to rotate about its point of connection with the radius rod. This has moved the valve forward, as indicated by the arrow, a distance equal to the lap of the valve plus the lead.

Considering diagram "1," Fig. 77; the main pin, eccentric crank, link and crosshead are in the same positions as they are in the corresponding diagram in Fig. 76. The link block, however, is nearer the center of the link. This has not affected the lead for the following reasons: The length of the radius rod is the same as the radius of the link, or approximately so. Thus, as the link is in its central position the raising of the link does not cause any movement of the front end of the radius bar. This point, which is the fulcrum of the lap and lead lever, is in the same position as in the

diagram previously considered. Consequently, this lever has moved the valve to the same position as before; and the lead is the same as when the reverse lever was in full gear.

In Figs. 78 and 79 the valves have inside admission. The radius rod is thus connected to the lap and lead lever above the valve stem.

The diagrams in Figs. 76 to 79 represent the various valve events throughout a complete revolution of wheels. Comparison between corresponding diagrams in the four figures brings out very clearly the difference in the arrangement of the Walschaert valve gear for outside and inside admission valves. It also shows the effect on the various valve events of hooking up the reverse lever. For example, it will be noticed that in Figs. 77 and 79 the valve cuts off the steam from the cylinders and opens the steam ports to the exhaust at a much earlier period in the stroke than it does in Figs. 76 and 78. The latter figures represent the engine with the reverse lever in the corner notch; while Figs. 77 and 79 represent running with a short cut-off.

With the eccentric crank a quarter of a revolution from the main pin, and with the eccentric of such length that the link is in its central position at the end of the stroke, the lead in the Walschaert valve gear is the same for all cut-offs.

To change the lead of the Walschaert valve gear, under the conditions given in the preceding paragraph, it is necessary to change either the lap of the valve or the distance between the connecting points of the lap and lead lever. Reducing the lap of the valve increases the lead, and vice versa. Changing the lap also changes the points of cut-off. If the lap is reduced, the cut-off will occur at a later period in the stroke; while if it is increased the opposite result will occur.

The following rules govern the changes to be made in the lengths of the arms of the lap and lead lever to increase or decrease the lead:

To *increase* the lead, make the upper arm (or distance between the valve stem and radius bar connections) longer in proportion to the lower arm (or distance between the radius rod connection and bottom connection of the lever).

To *decrease* the lead, make the upper arm shorter in proportion to the lower arm of the lever.

Arrangement.

A general outline of the gear for an inside-admission valve is shown in Fig. 80, where the names are given of all essential parts.

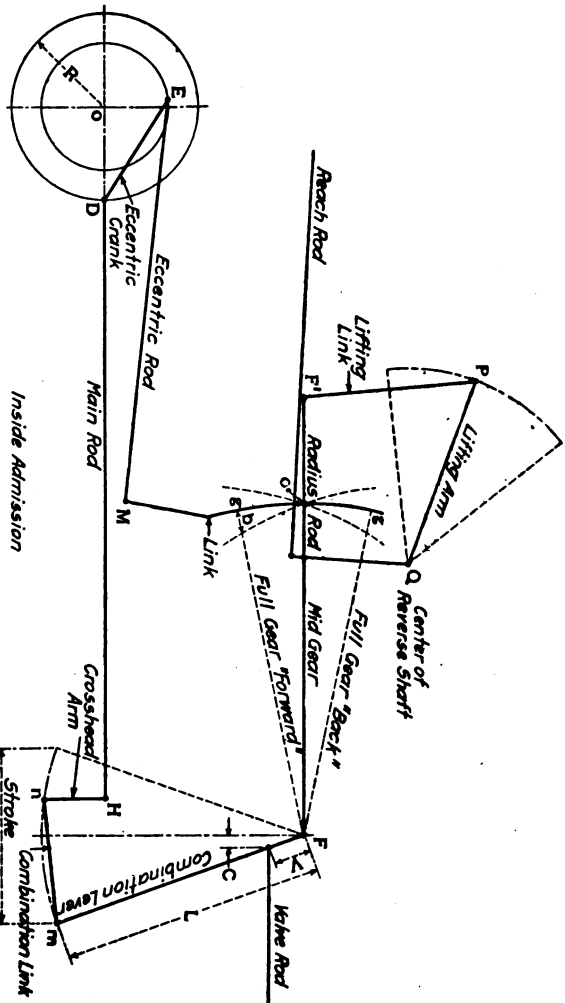

R
O
E
D
Eccentric Crank
Eccentric Rod
Main Rod
Inside Admission
Reach Rod
Lifting Link
P
F'
Lifting Arm
Radius Rod
C'
E'
B'
D'
Link
M
Full Gear "Forward"
Full Gear "Back"
Mid Gear
Q
Center of Reverse Shaft
Crosshead Arm
H
n
Stroke
Combination Lever
C
F
K
L
m
Combination Link
Valve Rod

FIG. 80.

The names define, as nearly as practicable, the special function each one has to perform in the combination of the gear. Starting at the crank pin "D" we have the eccentric crank, eccentric rod, link, radius rod, combination lever, combination link, crosshead arm and link block "g" (indicated only by letter). Fig. 81 is a similar arrangement of an outside-admission gear with no change in the names of any of the working parts.

It will be noted that in Fig. 80 the eccentric crank follows the main crank, while in Fig. 81 it leads. Also in Figs. 80 and 81 the radius bar is shown in the bottom portion of the link in forward motion, thereby reducing the stresses on the link trunnion bearings when running forward.

Before proceeding with the details of the gear, the stroke of the piston, lap and lead, virtual eccentric throw, valve travel and the point of cut-off must be shown. Of these the stroke, lead, valve travel and cut-off are determined with respect to the size of engine. It is advisable to add 1/32-inch to the actually desired lead to provide for wear of pins and bushings.

Lap.

When lead, cut-off and valve travel are given, the lap is found by the Reuleaux diagram, Fig. 82.

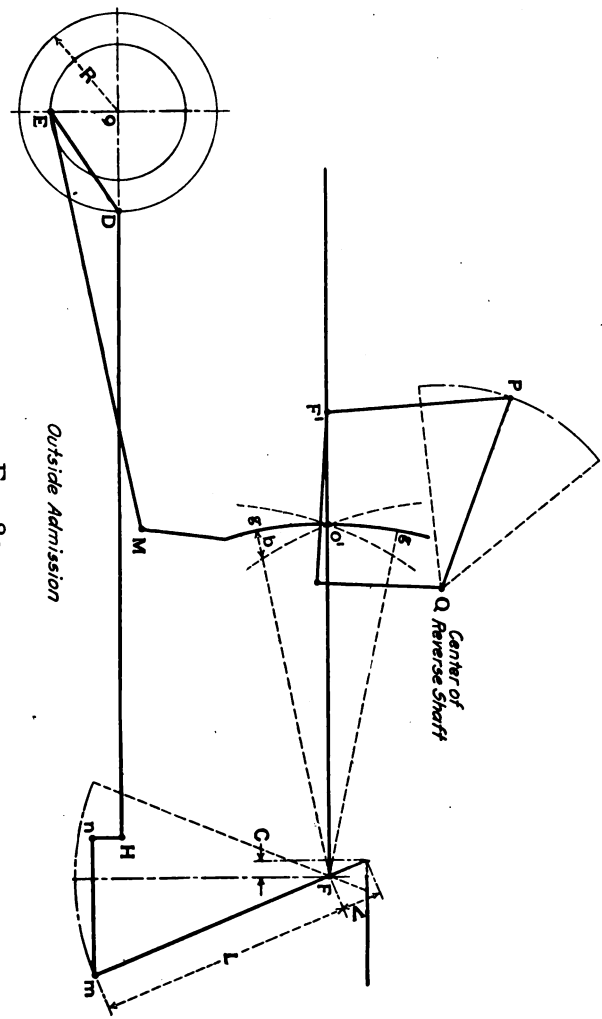

R
E
O
D
M
F'
O'
P
Q
Center of Reverse Shaft
Outside Admission
b
g
e
n
H
C
F
L
m

FIG. 81.

A circle with a diameter equal to the travel of the valve is drawn and on this diameter the same percentage as the desired cut-off percentage in the cylinder is measured from "A" to "B." From "B" is drawn

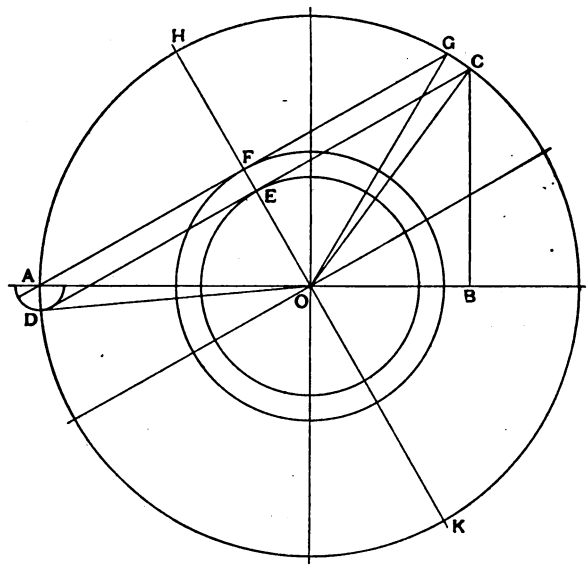


FIG. 82.

a perpendicular line which intersects the circle at "C." O-C will be the cut-off position of the crank. With the lead as radius and "A" as center, draw a semi-circle on the opposite side of the diameter to that of the location of point "C." From the cut-off point "C" draw a line tangent to the lead circle and extend same until

it intersects the valve circle at "D." O-D will be the position of the crank when the valve starts to open. Draw a line through "O" perpendicular to the line C-D, which it will intersect at "E." The distance E-O will be the required lap. The given lead may now be added to the lap directly by drawing a line through "A" parallel to line C-D which will intersect the extension of line E-O at "F." O-F will be the lap and lead, or the amount of motion to be imparted to the valve by the combination lever.

It will be noted that the angularity of the main and eccentric rods has some effect as to the exactness of the cut-off points in front and rear ends of the cylinder. This layout will, therefore, give an approximate average of what may be obtained in practice.

This method is equally applicable to the ordinary double eccentric motion with shifting links when the centers of the eccentrics are located at "A" and "G" with the crank at "K" for direct motion and outside-admission valve or for inside-admission when the motion is transmitted to same by means of a rocker. For inside-admission with direct motion or outside-admission using a rocker, the crank is located at "H" for the same eccentric position; provided, in all cases, that the center line of the cylinder passes through the center of the axle, otherwise their relation to the crank must

be changed to suit the angularity of the main rod when on the dead centers and the irregularity introduced thereby divided between the two dead centers. In the shifting link motion the link block will follow a convex curve from "A" to "C" in reversing from one full gear to the other, with increased lead toward central position with open rods, and a concave curve with diminishing lead toward center with crossed rods.

Returning to the Walschaert gear, we find that the combined motion of the single eccentric and the crank give an exactly equivalent result in full gear, but in linking up, the effect on the motion of the valve is equivalent to a link block following the straight line A-G from full forward to full backward gear, thus giving a constant lead E-F at all cut-offs.

Virtual Eccentric Throw—With Lap and Lead Known.

The required throw of point "F" (Figs. 80 and 81), being the final throw imparted by the crank E through the link to the combination lever, may be termed the virtual eccentric throw. It can be found either mathematically or geographically by the following formulæ, and as per Figs. 83 and 84, respectively. In both figures the valve circle is indicated by No. 2 and the virtual eccentric circle by No. 3.

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c}$$ for inside admission and

$$b = \frac{R\sqrt{a^2 + c^2}}{R + c}$$ for outside admission valves

where b = the radius of a circle with a diameter equal

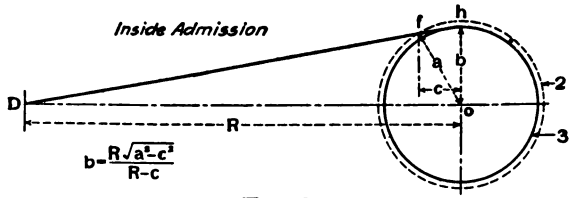


FIG. 83.

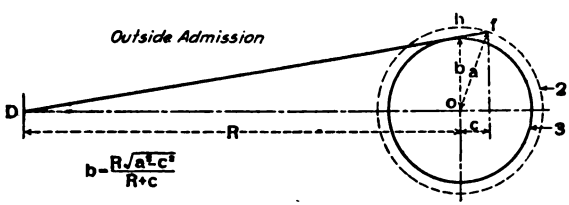


FIG. 84.

to the throw of point "f," (Figs. 80 and 81), representing the eccentric throw which may therefore be called the virtual eccentric throw.

R = the length of the crank = ½ piston stroke.

a = the radius of a circle whose diameter is equal to the valve travel.

c = the lap plus lead = distance O-F in Fig. 82.

Combination Lever.

It is evident that the motion of any point on the combination lever can be represented by a circle, as the two connections to which the motion is imparted, namely, "m" and "F" in Figs. 80 and 81 emanate from two circular paths, the main crank "D" and the eccentric pin "E" which is set at an angle of 90 degrees with the former, providing that the link connection of the eccentric rod is located on the center line of the engine in its central position. Even if the link connection is off the center line of the engine when the eccentrics will be more or less than 90 degrees in the lead or following of the crank, the effect of the movement of point "F" (Figs. 80 and 81) will be the same as if the eccentric crank had exactly 90 degrees angle to the main crank and consequently the line "b" in Figs. 83 and 84 will always be represented at right angles to the crank "D."

With "O" as center of axle, lay out the length "R" of crank "D." Through "O" draw a line perpendicular to length "R"; also with "O" as center draw the known valve circle.

For inside-admission valves, the lap and lead "C" is laid off from "O" toward the crank "P." From this point draw a line perpendicular to D-O until it inter-

sects the valve circle (No. 2) at "f." From "D" draw a line through "f" until it intersects the vertical diameter of the circle at "h," when O-h or "b" will be the radius of the vertical eccentric circle No. 3.

For outside-admission valves, Fig. 84, the lap plus lead is laid off on the opposite side of "O" to that of the crank and a perpendicular line is drawn in the same manner as in Fig. 83 until it intersects the valve circle at "f." Where the line D-f intersects the vertical center line of axle at "h," the radius "b" of the virtual eccentric circle is found by the distance O-h.

For outside-admission valve the eccentric circle will always fall inside of the valve circle; but for inside-admission the eccentric circle may fall inside of, on, or outside of the valve circle, depending on the relations between the stroke, valve travel and lap and lead.

The length of the combination lever will be made to suit the distance between the piston rod and valve stem, but its proportions will be V: L = C: R. The point of valve stem connection, measured from "m," will be L—V for inside, Fig. 80, and L+V for outside admission valves, Fig. 81, the valve stem connection falling below the radius bar connection in the former case, and above same in the latter case.

As a further illustration of how all the connecting points on the combination lever can be represented by

circles, refer to Fig. 85 for inside, and Fig. 86 for outside-admission valves which makes the whole system clear and complete. In these figures the swing of the

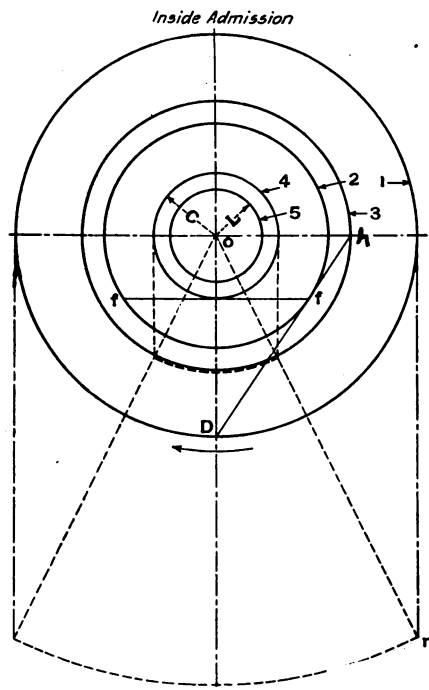


FIG. 85.

lower end of the combination lever is represented by the crank circle "1" from which it receives its motion. On the same center is drawn the valve circle "2," the virtual circle "3," the lap-plus-lead circle "4," and the

lap circle "5." Tangent to lap and lead circle "4" is drawn a line f-f perpendicular to the crank O-D until

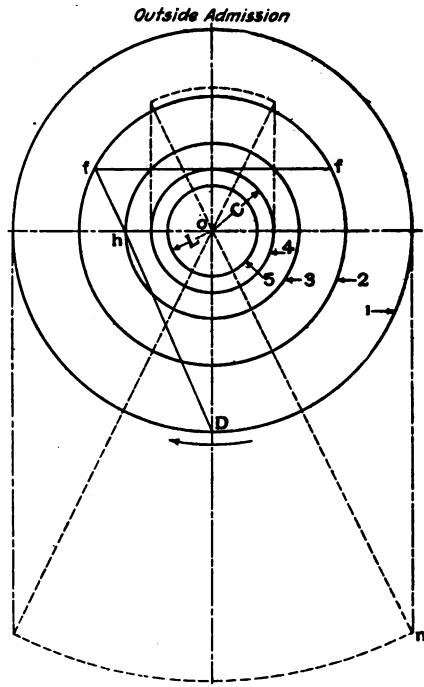


FIG. 86.

it intersects circle "2" at "f." By drawing the line D-f, extended to the transverse center line of the axle "O," it will intersect the eccentric circle at "h," which is in this respect a repetition of that found in Fig. 83

Fig. 86 gives the same arrangement, but for outside-admission, except that the line f-f is drawn tangent to the lap and lead circle on the opposite side of the axle "O" to that of the crank and the line D-F is drawn intersecting the transverse center line at "h," as in Fig. 84.

Length of Radius Rod.

With due consideration of the length of the eccentric rod, the radius rod should be as long as circumstances will allow. It is also subject to a minimum limitation in length, and a minimum of three times the travel of the link block "g" from full "forward" to full "back" gears is a good rule. See Figs. 80 and 81.

Link Slot Radius.

From the point of connection between the radius rod and the combination lever as center, strike off the link slot with a radius equal to the length of the radius rod.

Link Fulcrum.

At the same distance and on the same side above or below the extended center line of the valve stem as the average distance of the radius rod connection (point "F"), is the preferable location of the fulcrum of the link "O," Figs. 80 and 81. With a reasonably long radius rod a slight deviation from this will have little or no detrimental effect on the movement of the valve.

Link Swing.

The angular swing of the link should never exceed 45 degrees, preferably 40 degrees, from one extreme to the other.

Link Block Movement.

Full "forward" and full "back" positions of the link block "g," Figs. 80 and 81, will be at a point on the link where the swing is equal to the diameter of the virtual eccentric circle whose radius is "b," Figs. 83 and 84.

Throw of Eccentric Pin.

Determine the actual throw of the eccentric pin "E" by the relation of the location of the link block "g" in full gear, Figs. 80 and 81, to that of the eccentric rod connecting pin "M" from the link fulcrum. Generally the eccentric pin "E" will have two or three times greater throw than the virtual eccentric motions of points "g" and "F."

Length of Eccentric Rod.

Make the eccentric rod E-M as long as practicable, using three times the actual eccentric throw of pin "E" as a minimum length.

Eccentric Rod Connection to Link.

Special attention must be given to the location of the connection of the eccentric rod with the link pin at point "M" in Figs. 80, 81 and 87. This location must be plotted out because of the irregularity due to this

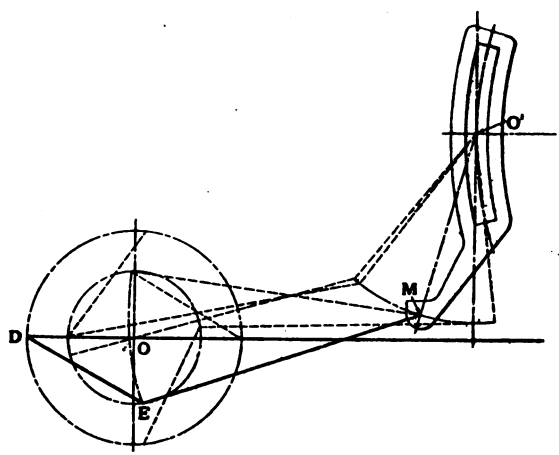


FIG. 87.

point "M" generally being some distance above the center line of the engine and also because of the angularity of the eccentric rod. The normal angle of 90 degrees between the crank "D" and the eccentric pin "E" must be modified to suit the height which the link pin "M" is above the center line of the engine and is determined in connection with the fore and aft locations of the pin "M." With limited lengths of eccentric rods, the

rear half of the eccentric circle gives a shorter throw than the front half. To compensate for this, the pin "M" will have to be moved back of the tangent of the link, until it gives, by the upward rise in its rear throw, the same angular swing to the link as that of the front from its central position when the crank is at its dead centers.

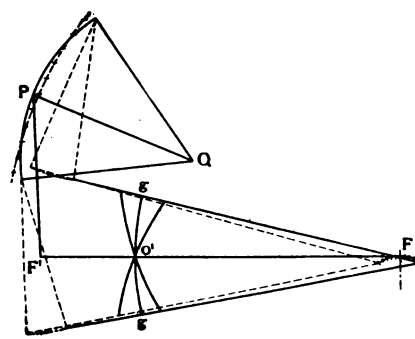


FIG. 88.

Great care should be exercised in locating this point so that the natural irregularities, unavoidable in the transmission of circular into lineal or oscillating motion, are properly compensated for in these particular points to get a correct and satisfactory motion of the gear as a whole.

Lifting Shaft Location.

The location of the lifting shaft "Q," Figs. 80 and 81, has an important bearing on the proper movement

of the link block "g" and consequently on the movement of the valve at all points between full forward and full backward positions. The lifting arm connection "P," Figs. 80 and 88, at upper end of the lifting link, should follow as near as possible a circular path described by the upper end of the lifting link in a position perpendicular to the radius bar while the link block "g" is moved from one extreme position to the other, with the main crank "D" at one of its dead centers, or being the same as a circle drawn with "F" as center and a radius equal to $\sqrt{s^2+u^2}$, where "s" is the length F'-F of the radius bar and "u" the length of the lifting link F'-P. The arm P-Q must necessarily be shorter than the radius thus obtained, and the lift link shaft "Q" should then be so located that with a convenient length of this arm, the point "P" will describe a curve intersecting the former at about 50 per cent cut-off position of the arm in both front and back gears. If, however, it is desired to favor the forward gear, a somewhat earlier intersection may be selected in the back motion, when the forward half of the curve will coincide more closely to the theoretical circle.

A somewhat closer refinement in this respect, is obtained by the location of the lifting shaft on the center line of motion back of link, Fig. 89.

The back end of the radius rod being raised and lowered by means of a sliding bearing, causes the link block "g" practically to follow the chord of the circular path described by the point in the link determining the throw of the block in all positions.

In the usual arrangement of the Walschaert valve gear, the lead is constant and does not vary as the reverse lever is hooked up. The Walschaert valve gear can, however, be so designed as to give a variable lead.

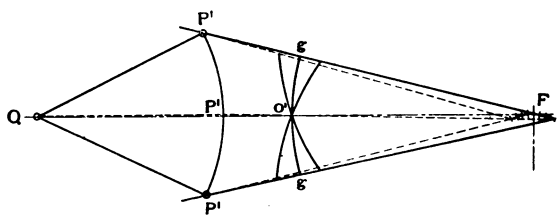


FIG. 89.

There are many cases in which the variable lead is preferred. With a variable lead, the longest possible cut-off in starting can be obtained, combined with the proper amount of lead at the ordinary running cut-off. In the case of passenger locomotives particularly, a steam distribution like this is often most desirable.

The favorable results for starting are, however, obtained at the expense of the distortion of the valve events in back motion. For this reason, the Walschaert valve gear with variable lead is suitable only for pas-

senger and fast freight locomotives; and not for slow freight and switching locomotives, and should therefore be used with discretion. This variation in lead is accomplished by designing the combination lever for the maximum lead in mid-gear and reducing it to a desired

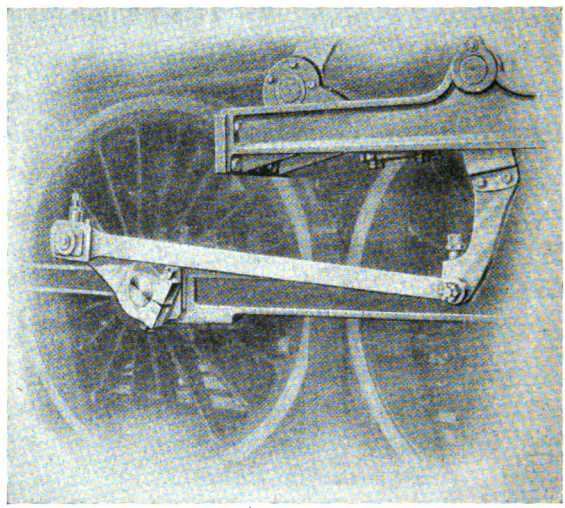


FIG. 90.

minimum at full gear by shifting the eccentric crank in the proper direction until this occurs.

The general arrangement of the Walschaert valve gear depends largely on the general design of the locomotive. Some of the ordinary forms of construction of the various parts of the gear are shown in the following illustrations.

Fig. 90 illustrates an eccentric crank and eccentric rod. The former is secured to an extension of the main crank pin by a binding bolt. The eccentric crank is split so that it can be drawn to a tight bearing by means of the bolt. This bolt, together with a key,

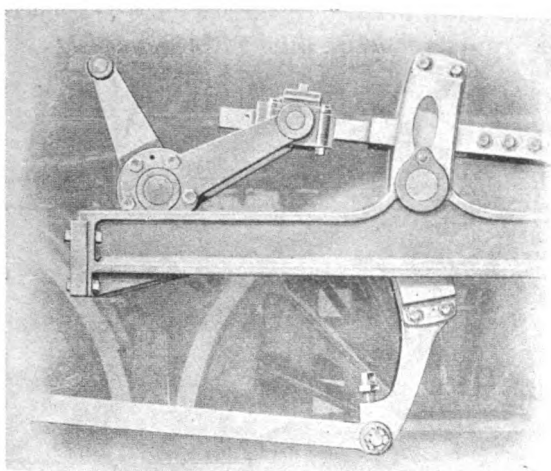


FIG. 91.

keeps the eccentric crank fixed in its position on the main pin. It may be easily removed, if necessary, by driving out the binding bolt. This construction has the advantage that it permits of using a solid bushing on the side rod at the main crank pin.

Fig. 91 illustrates the link and reverse shaft. In the design shown, the radius rod is directly connected to the lift shaft arm by means of a slip block. In such a case,

the reverse shaft arm is made in two parts, the outer part being easily removable. This permits of readily disconnecting the reverse shaft and radius rod and makes a very simple arrangement.

In another style of the reversing mechanism, also commonly used, the arm of the lift shaft is connected to

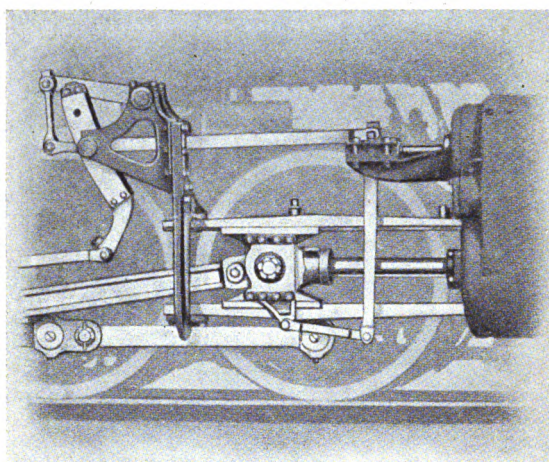


FIG. 92.

the radius rod by means of a link. This type of construction is shown in Fig. 92.

This figure also shows the American Locomotive Company's latest patented arrangement of self-centering guide for the valve stem of the Walschaert valve gear. It will be noticed that the guide is cast in one piece with the steam chest head. Consequently, the guide is

always central with the piston valve chamber. An arrangement of this kind has the advantage that it can be erected, taken down and replaced without any lining up. The guides are so constructed that they can be easily adjusted for wear by means of liners. This

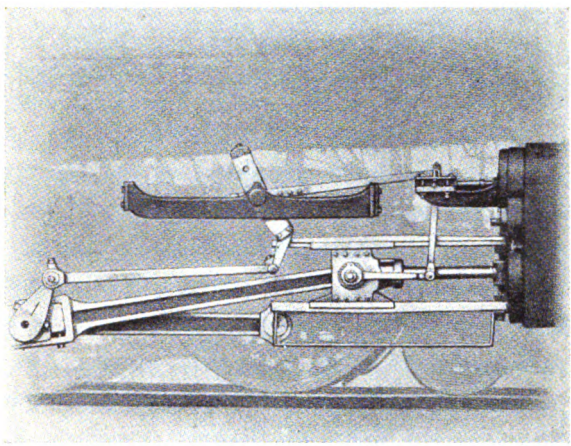


FIG. 93.

arrangement also permits the use of a straight design of lap and lead lever without forks. The lever is connected to the valve stem crosshead by a pin passing through its wings. In case it is necessary to take down the lap and lead lever, the upper portions of the guide may be removed by taking out four bolts. This gives access to the pin connecting the lap and lead lever to the crosshead.

Fig. 93 illustrates the latest arrangement of the combination link directly connected to the wrist pin. This arrangement has the advantage of eliminating the crosshead arm and bushing, thereby reducing the number of parts and also reducing the reciprocating weights. It also simplifies the construction of the guide yoke end.

However, on long stroke engines, the pins are so close together that a good arrangement is not practical.

In Case of Accidents.

Having considered the principles and construction of the Walschaert valve gear, the next phase of the subject to take up is what should be done in case of accidents. This is of the utmost interest to all engineers. It would be impossible to lay down rules to cover every case that might arise. It is possible, however, to consider some of the more usual or most possible accidents; and determine the best and quickest courses to follow in such cases.

The accidents to be considered may be divided into two general classes; those in which it is not necessary and those in which it is necessary to block the valve to cover the ports.

The first class of accidents includes only those cases in which it is necessary to take down the main rod. The valve must necessarily be blocked if the main rod is disconnected.

In the second class of accidents, the damage to the machinery may or may not make it necessary to take down the main rod.

Main Rod Up, Valve Not Blocked.

The following is considered a good rule to follow in regard to taking down the main rod. When it is possible to lubricate the cylinder and relieve compression, otherwise than by removing the cylinder cocks, the main rod may be left up if in a condition to run. For example: if there are relief or vacuum valves in the cylinder heads, these may be removed. This will prevent compression and also permit of lubricating the cylinder.

Considering the first of the above mentioned classes of accidents in which the valve does not have to be blocked; suppose, for example, an eccentric crank, eccentric rod or the foot of the link is broken.

Fig. 94 illustrates how the valve gear may be disconnected in such cases. Take down the eccentric rod, disconnect the radius rod from the lift shaft and secure the link block in the center of the link. The valve on the lame side then receives a motion from the lap and lead lever. Its travel will be equal to twice the total amount of the lap plus the lead. This gives a port opening equal to the amount of the lead. Consequently, the main rod may be left up, as the cylinders will be lubricated. Furthermore, though the cut-off on the

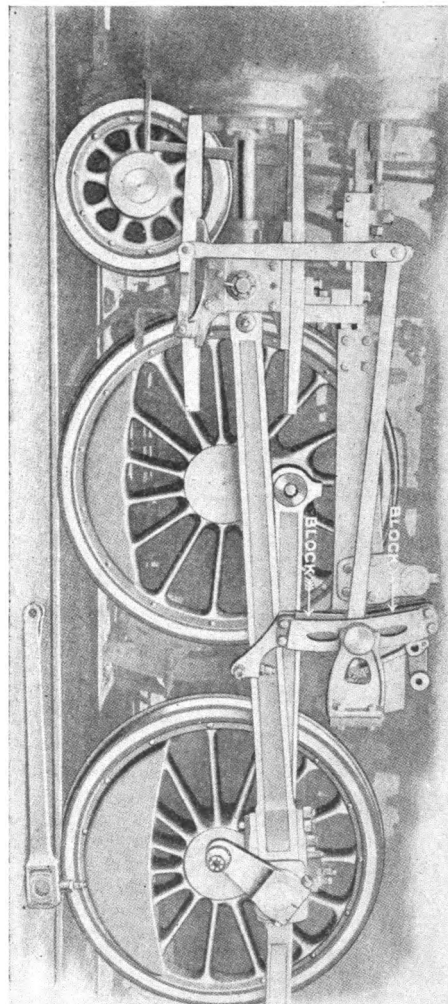


FIG. 94.

disabled side will be very short, the steam that is admitted will do a certain amount of work, and the engine can be reversed.

The link block may be secured in the center of the link by means of two blocks wedged in position, as shown in the illustration.

With the link block thus secured, care must be taken not to come to a stop with the main pin on the lame side on either quarter. In such an event, the lap and lead lever would be in a perpendicular position, or approximately so; and the valve would be practically central on its seat. As the crank pin on the other side would be on a dead center, it would, thus, be impossible to start the locomotive.

In Fig. 94 the radius rod is connected to the lift shaft arm by means of a link or hanger. It can thus be readily disconnected.

When the radius rod is directly connected to the lift shaft arm by means of a slip block as shown in Fig. 95, the reverse shaft arm is made in two pieces. To disconnect the radius rod, it is necessary only to remove the outer section of the lift shaft arm and remove the slip block.

Considering the second general class of accidents: those in which the valve has to be blocked to cover the ports. This class, as already stated, may in turn be

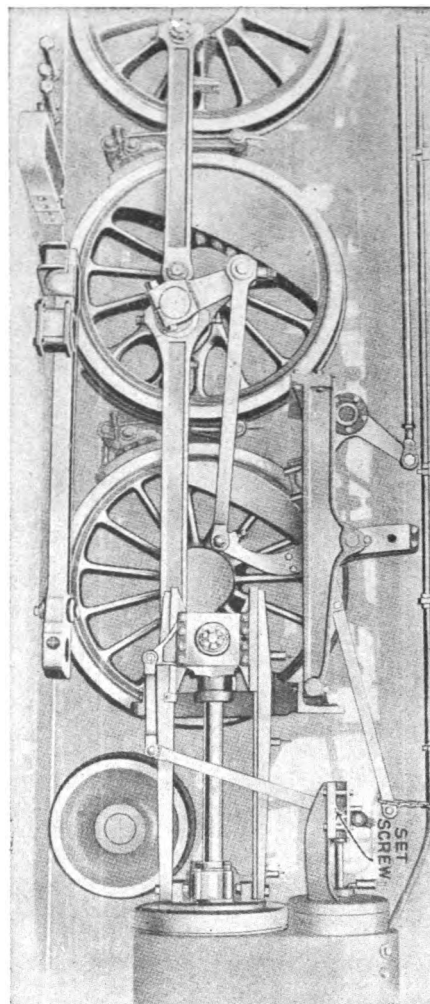


FIG. 95.

divided into two other classes: First, cases where it is necessary to take down the main rod; second, those in which the main rod is in condition to run. In the latter cases, the rules governing what is considered good practice as to leaving the main road up or taking it down has been previously given.

Taking up these two kinds of accidents in their order:

Main Rod Down, Valve Blocked.

Assume that the main rod is broken or the piston rod bent. In the case of the inside-admission valves, the locomotive may be disconnected and blocked as shown in Fig. 95. Disconnect the radius rod from the lap and lead lever. Suspend it clear of the latter. Secure the valve to cover the ports. This can usually be done by means of a set screw provided for this purpose. In the design illustrated, the set screw will be noticed on the side of the valve stem guide. If no set screw is provided, a valve stem clamp is usually included in the tool equipment. Clamp or block the crosshead at the back end of the guides as shown.

With the valve motion disconnected in this way, the reverse lever is free to operate the other side; and the locomotive can be run in on one cylinder.

If the crosshead arm, lap and lead lever connector, or lap and lead lever is broken, the same method may be followed. Of course, such of the broken parts as

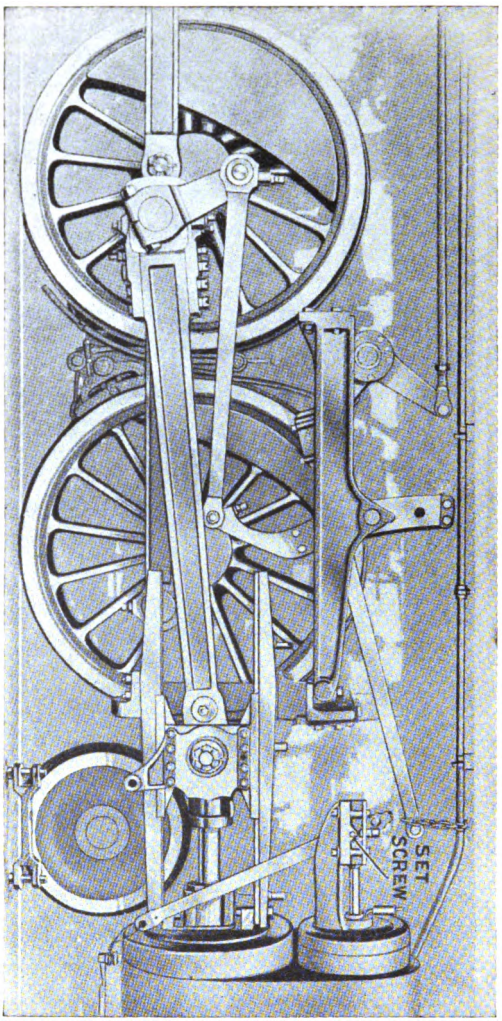


FIG. 96.

would in any way interfere with running the locomotive would have to be removed.

In the case of *outside* admission valves, the same course may be followed as illustrated in Fig. 95, and described above; except that the lap and lead lever must be taken down. If this was left up, the radius rod would strike it as the latter moves back and forth to the motion of the link.

Main Rod Up, Valve Blocked.

In such a case, if the valve has *inside* admission, the locomotive might be blocked as shown in Fig. 96. For example: suppose the lap and lead lever, lap and lead lever connector, or crosshead arm was broken. Disconnect the radius rod from the lap and lead lever. Suspend it clear of the latter. Tie the lower end of the lap and lead lever ahead to clear the crosshead on the forward stroke. The locomotive can then be run in on one side.

If the valves have *outside* admission: under conditions similar to those assumed in the case of Fig. 96, the lap and lead lever must be taken down. In other respects, the same method may be followed.

When the radius rod is suspended, a chain or wire should be used. A rope would be quickly cut through by the motion of the radius rod.

Setting the Walschaert Valve Gear.

If the gear is properly designed, the setting of the Walschaert valve motion is very simple.

It is essential that the length of the eccentric crank, the lengths of the arms of the lap and lead lever and the length of the lap and lead connector should check exactly with the drawings. No change from the drawings in any of the above parts should be made.

Assuming that all the parts check with the drawings, any adjustments which may be necessary to square the motion should be made in the lengths of the eccentric rod, radius bar or valve stem.

Diagrams "A" and "B," Fig. 97, illustrate, respectively, the effects on the position of the valve resulting from changes in the lengths of the radius rod and eccentric rod.

It is apparent from diagram "A" that any change in the length of the radius rod "R" will have a corresponding and practically equal result on the position of the valve. That is, if the radius rod is lengthened ⅛-inch the valve will be moved forward on its seat a like amount, or vice versa. Therefore, to move the valve ahead or back any amount, lengthen or shorten the radius bar respectively a like amount.

Reference to diagram "B," Fig. 97, shows that the effect of a change in the length of the eccentric rod

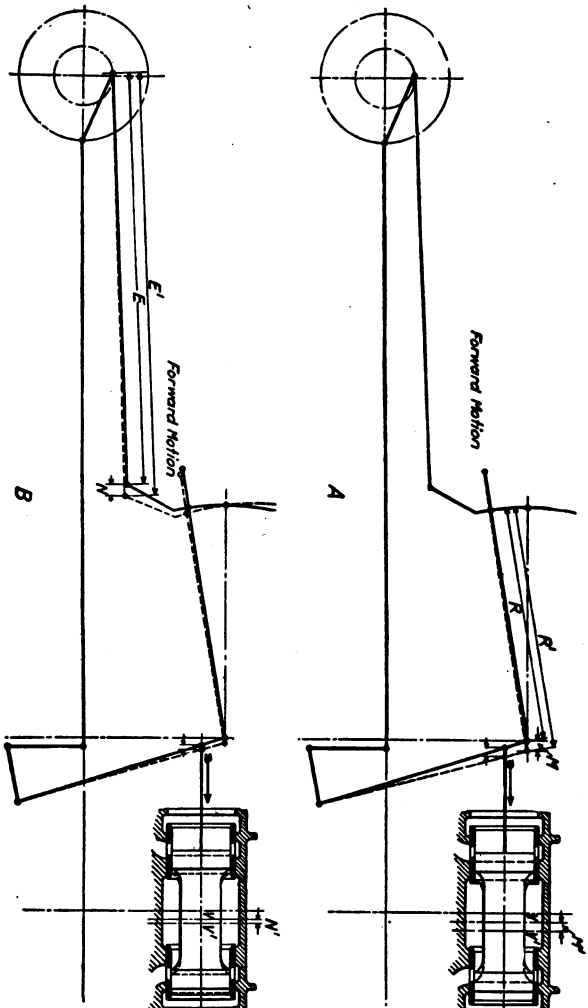


FIG. 97.

on the position of the valve depends on whether the link block is in the top or bottom of the link. The following rules therefore govern:

When forward motion is taken from the *lower half* of the link.

In *forward motion:* To move the valve ahead, lengthen the eccentric rod. To move it back, shorten the eccentric rod.

In *backward motion:* To move the valve ahead, shorten the eccentric rod. To move it back, lengthen the eccentric rod.

When forward motion is taken from the *upper half* of the link.

In *forward motion:* To move the valve back, lengthen the eccentric rod. To move it ahead, shorten the eccentric rod.

In *backward motion:* To move the valve ahead, lengthen the eccentric rod. To move it back, shorten the eccentric rod.

A change in the length of the eccentric rod will not alter the position of the valve a like amount. The two will have approximately the same ratio to each other as that between the eccentric crank throw and the valve travel.

For example: Suppose the eccentric throw is 12 inches and the valve travel 6 inches. The ratio is two

to one or the latter is one-half of the former. In such a case, a change in the length of the eccentric rod of ⅛-inch would alter the position of the valve only one-half of that amount, or 1/16-inch.

The following method of setting the Walschaert valve gear is equally applicable to designs having constant lead and those having variable lead.

First: Check the lengths of the eccentric crank, the lap and lead lever, and the lap and lead lever connector. These dimensions should conform to the drawings and should not be altered.

Second: Raise the main wheels so that the distance from the center of the wheel to the top of the frame conforms to the amount specified, minus ½-inch for wear of brasses. Then find the dead centers and port marks in the usual manner.

Third: Assemble the gear complete, temporarily tightening the eccentric crank in a position which will give the specified throw.

Fourth: Place the radius bar in the center of the link. Mark the mid-gear position. Then get the lead at each end of the cylinder. When the lead is constant, the *average* lead, or the sum of the leads on opposite ends divided by two, should be equal to the specified lead in full gear. When the lead is variable, the *average* lead in mid-gear position should be equal to one-half of

the sum of the specified leads in full forward and full back gear. In other words, it should be the lead due to the lap and lead lever, unaffected by the position of the eccentric crank. Any error in the average lead when the radius bar is in central position is due to an error in the length of the upper or lower arms of the lap and lead lever.

Having thus checked the lengths of the lap and lead lever, equalize the lead by means of the adjusting nuts on the valve stem or by changing the length of the radius bar.

Fifth: Drop the lever into forward gear until the specified travel is obtained. Then, if the average lead is equal to the specified lead in full gear, the eccentric crank is correctly set. If this is not the case, the eccentric crank should be driven one way or the other until the error is corrected. If the average lead is less than the specified lead, the eccentric crank should be driven *inward,* if it *leads* the main pin; and driven *outward,* if it *follows* the main pin. If the average lead is more than the specified lead, the eccentric crank should be driven *outward,* if it *leads* the main pin; and driven *inward* if it *follows* the main pin.

After eccentric crank is correctly set, check valve travel; and relocate full forward position of the reverse lever.

Sixth: If the average lead is correct but unequally divided on the front and back centers, lengthen or shorten the eccentric rod, according to the rules given above, until it is equalized. It must be borne in mind that to change the lead a given amount the eccentric rod must be changed a greater amount, or about in proportion as the eccentric throw is greater than the valve travel. (See Fig. 97.)

Place the reverse lever in a position that will give full travel to back gear, marking this position on the quadrant; and check the lead in the same manner. With variable lead, the full back gear lead should be as much greater than the lead at mid-gear as the lead at mid-gear is greater than that at full forward gear.

Seventh: Run over the cut-offs and obtain other events for as many positions as required. In running over the cut-offs of locomotives of the articulated type, obtain the cut-offs for each position of the lever for *both engines* before moving the lever to a new position. This is necessary in order that the relative cut-offs in high and low pressure cylinders may be compared. *Note:* Do not attempt to square cut-off at the expense of lead and port opening.

Variable Lead.

When this setting is resorted to, it is advisable to adjust the gear according to the general rule, using a

temporarily fixed eccentric crank to obtain the proper length of the eccentric rod and valve location, afterwards, readjust the eccentric crank to suit the desired lead in full gear.

The Kingan-Ripken Valve Gear Device.

A simple and practical device, designed to overcome the slow action the combination lever imparts to the Walschaert, and similar valve gears, was patented April 16, 1912, by James B. Kingan and Hugo F. Ripken, of Minneapolis, Minn.

This device is the result of considerable study and experiment in an effort to overcome the slow starting of trains, and maintenance of speed, by engines equipped with the Walschaert valve gear, which is due to the heavy constant lead required at short cut-offs, in order to maintain sufficient port opening, where little opening other than that transmitted by the combination lever is given to the valve. This constant lead is too excessive for the maximum cut-off, and the starting of a train, and, as it requires one-half the movement of the crosshead in order to overcome the lap and lead, has a tendency to slow the action of the valve gear, as ordinarily connected.

This arm, in addition to receiving the usual crosshead motion, also receives a supplementary motion from

the main rod, for now being attached directly to the main rod, it receives the same oscillating movement the rod does from its pivot, the wrist pin.

And the fact that the largest part of this movement is transmitted to the combination lever as the crosshead approaches and recedes from the end of its travel, causes it to give to the combination lever an entirely different movement from that transmitted by the usual crosshead connection.

This movement, as shown in the illustration, causes (as the piston is on the last half of its stroke) release, compression, and pre-admission to occur later in the stroke with the same cut-off, thereby giving a longer period of expansion, less compression, and through the fact that the supplementary motion transmitted is the greatest just as the crosshead approaches and recedes from the end of its travel, there is a great deal less pre-admission with the same amount of lead. The piston having reached the end of its travel, the main rod is now in the center of its oscillatory movement and the device has therefore no effect upon the lead. But as the piston leaves the end of its travel and the main rod again assumes an angle, a quicker, larger and longer maintained port opening is obtained with the same cut-off and also a longer cut-off with the reverse lever in the same notch, or position. By placing the combina-

tion link connection in this device in such a position as to take advantage of the various angles that may be transmitted to the combination link, through its use a much more even distribution of steam is obtained in the cylinders.

Later release and compression, less pre-admission, a quicker, larger and longer maintained port opening with

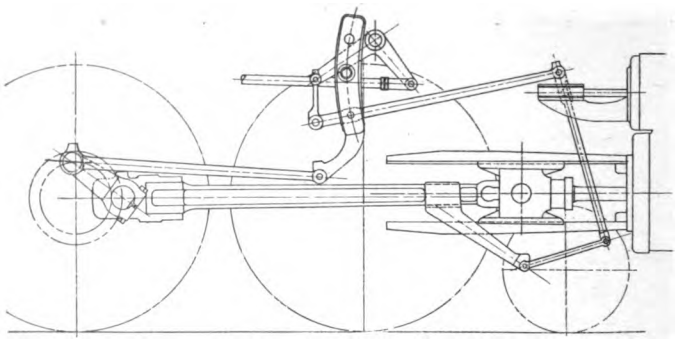


FIG. 98.

the same cut-off, and more uniform valve events, allow the use of a shorter cut-off wth its constant saving of steam, and this fact, combined with the longer maximum cut-off obtainable, makes it absolutely certain that a quicker starting, faster, more powerful and more economical engine is obtained.

DESCRIPTION OF OPERATION.

A full conception of the device may be had by reference to the accompanying illustration. When the cross-

head is approaching the forward center and the main rod is in an inclined position, the arm being rigidly attached to the main rod, its bottom end, or union link connection, is the same degree back of the pivot point (the wrist pin) as the main rod is above it. The combination lever, being connected by means of the union link to the bottom end of the arm, has its action delayed in the same proportion.

The events of release, compression and pre-admission, taking place in this part of the stroke, are delayed proportionally with the delayed action of the combination lever.

Referring to Fig. 98 it will be observed that the crosshead having reached the end of its travel in this position, the main rod is in the center of the oscillatory movement derived from the wrist pin, and no movement being transmitted to the combination lever other than that given by the crosshead, the combination lever is therefore in the same position it would occupy if connected directly to the crosshead. Lead is therefore not affected by the use of this device.

Following the position shown in Fig. 98, the crosshead commences its return stroke, and the main rod, having passed its pivotical center, is again in an inclined position, and has moved the bottom end of the arm ahead of the position it held in relation to the crosshead while

on center, and in so doing has delayed the return movement of the combination lever.

The combination lever traveling in a direction opposite to that of the radius rod which is opening the valve port in this position, would greatly tend to slow the action of the valve gear. But this device, in delaying the return movement of the combination lever, gives to the valve a quicker, larger and longer maintained port opening with the same cut-off, and a longer cut-off with the reverse lever in the same position, than the Walschaert or any other valve gear using the combination lever can give, with the old style crosshead connection.

The invention is marketed by Mudge & Company, Chicago, Ill.

Davis Variable Lead Attachment.

The device herewith presented for consideration was patented December 6, 1910, by Mr. William Lynn Davis, to whom we are indebted for the following description:

"For many years there has been a desire of locomotive designers to obtain a variable lead valve gear showing a steam distribution equally as good as the Stephenson but which would embody in its design the accessability and the simplicity of the Walschaert.

In consideration of this, the valve gear embodying the Davis variable lead attachment was designed, as shown in the accompanying illustration, to give a variable lead

increasing as the link block travels from the ends of the link toward the center, or, as the engine is "hooked-up," and vice versa, giving the valve gear its proper and most efficient movement at the beginning of the stroke both in full gear and in the cut-off, thus making it especially beneficial in starting on account of leaving the engine free of all preadmission, lengthening out the cut-off, and also delaying the exhaust. In other words, it has been demonstrated that an engine with the Davis attachment whose cut-off in full gear before the attachment was applied was 21 inches, was 23½ inches after the attachment was applied, showing an increase of 2½ inches in the distance that the steam followed the piston, and delaying the exhaust occurrence when the piston was within 1⅛ inches of the completed stroke; this, of course, aiding greatly in starting the engine. By means of this attachment any desired lead can be obtained at two points, in the cut-off and at full gear, without any change in design.

Further: In the Walschaert type of valve gear the location of the pivot connection joining the lap and lead lever and the radius rod determines the amount of valve lead, and it is a difficult operation to locate this pivot at the most desirable point, frequently involving many expensive trials and requiring the reconstruction of the lever or the making of new ones. When the most de-

sirable location has been found, to suit the usual conditions arising in the running of a locomotive, the distribution of the steam is still faulty and ineconomical, the best results in steam engine practice being obtainable only by varying the lead in relation to the cut-off so that as the main stroke of the valve decreases, the lead increases, and vice versa. In the Stephenson link motion the variation of the lead is accomplished by the combination of the motions received from the two eccentrics set at relatively different positions to the crank pin and the rotation of the strap and link around the eccentrics as the link is raised or lowered. The principal advantage of the Davis attachment is to provide means for easily varying the relative lengths of the arms of the lead and lap or combination lever, with the Walschaert type of valve gear, whereby the valve lead may be easily varied. In making the pivot joint movable its location may be easily and readily altered or changed through a range which will vary the lead from the greatest to the least desirable amount under the conditions existing at different times. Now if this change can be accomplished automatically without interfering with or adding to the duties of the engineer, the principal object has been obtained.

In designing the valve gear illustrated here it will be noticed that the distance between the two upper pins in

the combination lever is subject to change through a U hanger and a bell crank, the latter being connected to the hanger pin on the tumbling shaft by the main connecting rod, or, as it has been named, the lead and lap adjusting rod.

In other words, the main advantage of the Davis attachment for the Walschaert and kindred types of

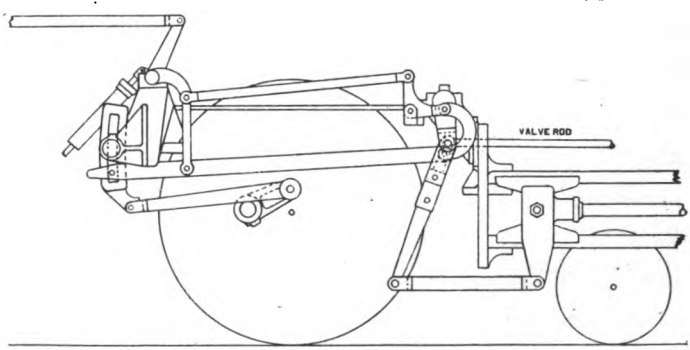


FIG. 99.

valve gears resides in the combination of the lead and lap lever and the radius rod with the shiftable fulcrum pin, by which the lead and lap lever and radius rod are joined, and means for shifting this fulcrum in order to automatically vary the action of that part of the valve mechanism which produces the lead, when the reverse lever is moved to adjust the cut-off. Fig. 99 simply illustrates the valve gear as designed to cover the Atlantic type of balanced compound passenger engine in use since

1912, on the Atchison, Topeka & Santa Fe Railway. It is readily understood that the details, in order to be applicable to other valve gears of this type than the Walschaert, can be modified or altered to accommodate the construction of that particular valve gear. From the accompanying illustration it will be noticed that the valve gear in general is of the ordinary Walschaert type, except the upper end of the combination lever, which is bifurcated, and the front end of the radius rod, which, instead of being pinned directly to the combination lever, has trunnions on its front end which carry blocks, the blocks in turn sliding in the jaws provided on the inside faces of the combination lever. The combination lever carries on its outer faces bosses which engage in the rocker arm, which, in turn, moves the valve. It is readily seen that if the engine were "hooked-up" from the position in which it is shown in the illustration, the upper radius rod hanger pin would come more nearly in a line with the center of the tumbling shaft and the upper end of the bell crank, which would tip the bell crank forward, thereby pushing down the U hanger and with it the trunnioned end of the radius rod, increasing the distance between the centers of the pivot connection on the radius rod and the rocker arm connection, thus increasing the lead.

It has been demonstrated time and again that the Walschaert valve gear makes an engine a poor accelerator and engines equipped with that gear have given trouble in starting trains; especially heavy trains which require the maximum tractive effort; also local passenger trains where rapid acceleration is of prime importance. The application of the Davis attachment is low in cost and it does not in any way destroy the accessibility feature of the Walschaert gear. It adds little to the complexity and will make any Walschaert valve gear more economical and flexible and will make an engine employing it a far better accelerator."

Mr. Davis is at present employed as a Mechanical Engineer by the Interstate Commerce Commission, at Kansas City, Mo.

THE BAKER VALVE GEAR.

The original Baker valve gear for locomotives was the joint invention of Mr. Abner D. Baker, of Swanton, Ohio, and Mr. Charles J. Pilliod, of Angola, Ind., and an American patent on the valve gear was issued November 3, 1908.

This form of valve gear was primarily designed for traction engines, and was originally known as the Baker-Pilliod valve gear, but during 1908 it was first applied to a locomotive and passed through the experimental

state, or period, of development covering about two years.

In 1910 the gear was entirely remodeled, and some organic changes were made in its mechanical construction, which increased its efficiency and stability to a considerable extent. As a whole, the gear is now more accessible and compact, weighs less, and can be more easily applied.

The driving force of this valve gear is, like the Walschaert, derived partly from the eccentric crank and partly from the crosshead. In this respect the two gears are similar. The most important distinction between the two types is the absence of a fixed, or shifting, radial link in the Baker, as may be seen in Figs. 103 and 104, and its employment in the Walschaert. The reverse bars and reverse yoke, used in the Baker valve gear, take the place of the link used in the Walschaert and Stephenson valve gears.

The combination lever of both types of gear move the valve the amount of lap and lead in each direction, while the balance of the valve travel is obtained from the eccentric crank. In the Baker gear the eccentric crank transmits its motion through the eccentric rod to the gear connecting rod, radius bar, bell crank and valve rod, dispensing with the oscillating radial link, link block, and radius bar, employed in the Walschaert gear,

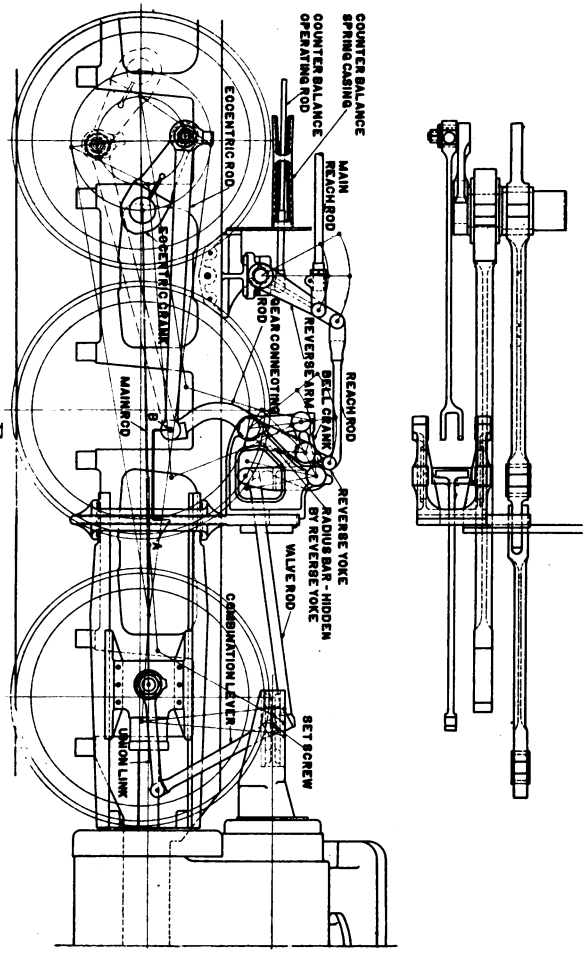
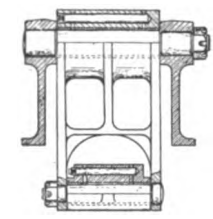

COUNTER BALANCE SPRING CASING
COUNTER BALANCE OPERATING ROD
MAIN REACH ROD
ECCENTRIC ROD
ECCENTRIC CRANK
MAIN ROD
REVERSE ARM
REACH ROD
BELL CRANK
REVERSE YOKE
RADIUS BAR - HIDDEN BY REVERSE YOKE
VALVE ROD
COMBINATION LEVER
SET SCREW
UNION LINK

FIG. 100.

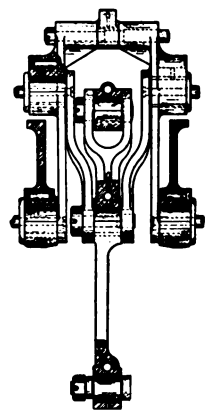
and thus avoids the usual variation occasioned by the slip of the link block.

Description.

The Baker valve gear is an outside gear having no links or eccentrics. The bearings of all moving parts



SECTION AT BELL CRANK



SECTION AT REVERSE YOKE

FIG. 101.

are pins and bushings, making it easy to repair. The movement of the valve is controlled by the reverse yoke, shown in Figs. 101 and 102, which is pivoted in the gear frame. This reverse yoke carries the radius bar, on the lower end of which is pivoted the gear connecting rod. This gear connecting rod extends from the radius bar bearing downward to connect with the eccentric rod and upward to connect with the bell crank.

The minimum vertical movement of the gear connecting rod is obtained with the reverse yoke in mid-gear, and, as the reverse yoke is moved toward either extremity, the valve travel is increased through the increased oscillation of the bell crank, by means of a greater ver-

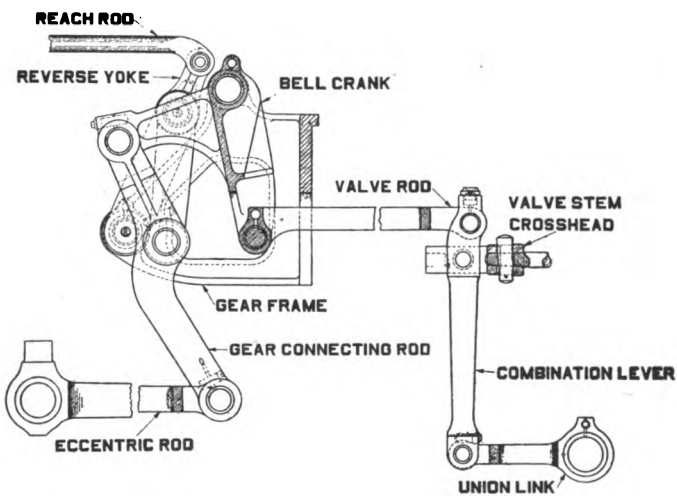


FIG. 102.

tical movement of the three gear connecting rod bearings.

A movement of the lower end of the gear connecting rod, Fig. 102, to the left, with the reverse yoke in forward motion, produces a lifting action of the gear connecting rod, which moves the valve back by revolving the bell crank clock-wise. This same movement of

the lower end of the gear connecting rod will produce a falling action of the gear connecting rod with the reverse yoke in the back motion, which reverses the movement of the bell crank and valve. The crosshead connection moves the valve the amount of lap plus lead in each direction. This makes the lead constant, and independent of the cut-off.

Application.

This valve gear may readily be applied to all modern equipment, either old or new, and to any class of service. There is but one type of gear for inside steam distribution, shown in Fig. 103, and one for outside steam admission locomotives, illustrated in Fig. 104. They are applicable and interchangeable for all classes of service.

ADVANTAGES.

The Baker, like other outside valve gears, has numerous advantages over inside valve gears. Being located outside, it relieves the complexity and congestion underneath the engine and permits practical bracing of the locomotive frames. The gear is standard in all parts, regardless of size or class of engine.

After setting, the valve gear keeps its adjustment permanently. Having no links, eccentrics, or sliding blocks, there is no chance for variance, as there is in the ordinary link gear.

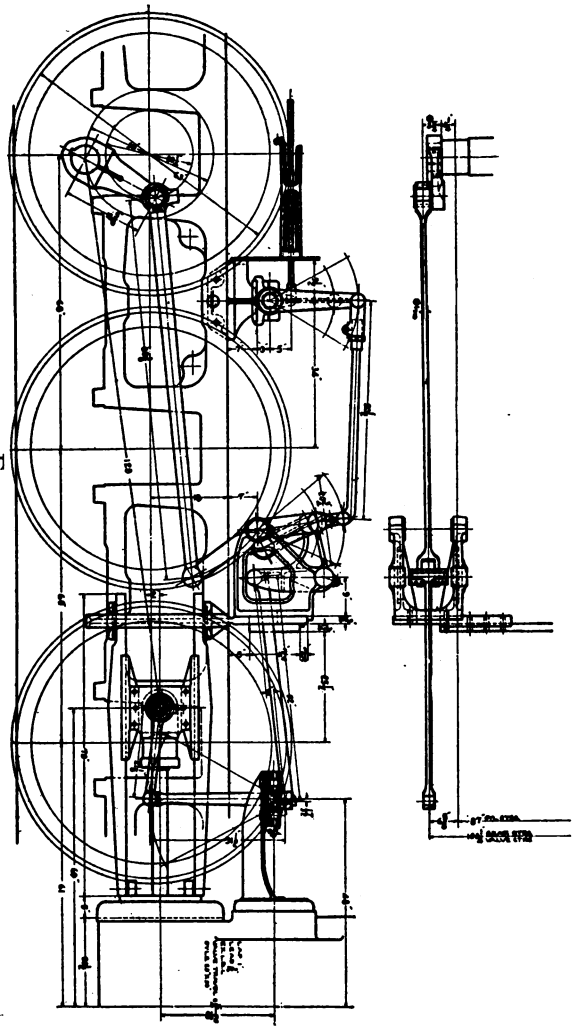


FIG. 103.

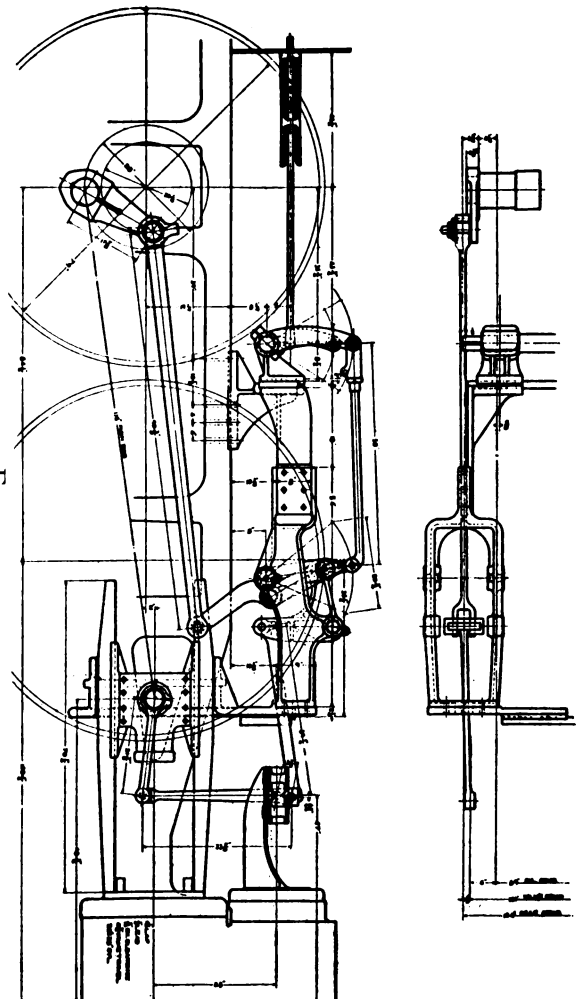


FIG. 104.

All parts of the gear are easily accessible, making inspection easy and renewals when necessary after long service, inexpensively and quickly made. Every moving part is equipped with an integral oil well, thoroughly protected from dust and dirt, insuring positive lubrication. The points where wear occurs are all bushings and pins, very quickly and easily replaced, bringing the gear into its original alignment and proper relationship with the valve. This is not obtainable with some other gears where bushing points are independent of the locomotive, or where wear or distortion of the link, or the block, will disturb the steam distribution. A more uniform cut-off and release is obtained and therefore greater effective expansion from the steam.

Few repair parts must be carried in stock, for several parts will fit either right or left or both inside and outside admission gears, and most of the other parts are interchangeable right and left. In making replacement of worn or damaged parts on the Baker gear, all that is necessary is to remove a couple of nuts and pins, slip the stock part in place, and the engine is again ready for service.

Setting the Baker Valve Gear.

Assemble the valve gear in accordance with the design furnished. Valve travel, lap, lead and exhaust clearance are shown on assembling design.

It is essential that the length of the eccentric crank, combination lever arms, union link, and valve stem check with drawings.

Clamp the eccentric crank temporarily to the main pin, and locate as near as possible to the specified throw. Locate the dead centers in the usual way. Place the reverse lever in full forward motion position, while locating the dead centers. In doing this full travel, lead, and all clearance can be tested, and, in a great many instances, sufficient information can be obtained from the marks that have been made on the valve stem from original adjustment as will enable the valve setter to make alterations with one complete revolution of the wheels.

Eccentric Crank Setting.

With the engine on the front dead center, tram from the center of the pin in the front end of the eccentric rod to any stationary point, such as the guide yoke or guides, as shown by the tram points 'A' and 'B' in Fig. 100. (In most cases the wheel tram can be used for this work.) After scribing a line across the side of the main guide or guide yoke end with the 'A' end of the tram, revolve the wheel to the back dead center, and scribe the guide or guide yoke end again; if these two lines are together the crank setting is correct. If they are not, move the eccentric crank in the required

direction until these lines come together. The position of the reverse lever is not important while finding the eccentric positions. The eccentric crank always follows the main crank pin.

Eccentric Rod Length.

The inside admission valve gear is direct in forward motion and indirect in back motion. The ratio of the valve gear is about 4 to 1; therefore, the valve will move forward 1/16 inch if the eccentric rod is lengthened ¼ inch, with the reverse lever in extreme front stroke position and the crank pin on front dead center. With the reverse lever in extreme back stroke position and the crank pin on back dead center, the valve will move back 1/16 inch when the eccentric rod is lengthened ¼ inch. Having taken the port marks with standard valve stem tram, the object is to square the lead in forward and backward motion on front and back dead centers.

Changes necessary on the eccentric rod and the valve rod are determined from lead marks.

Let us take a valve with ¼ inch lead, showing a difference between front and back motions of ⅛ inch, making it necessary to move the valve backward 1/16 inch to square the forward motion lead with the above ratio, 4 to 1, the eccentric rod must be shortened ¼ inch. On inside admission gears the direction of the change in the eccentric rod should be the same as that

in which the valve should move to correct the forward motion.

Do not try to square cut-offs by squaring the port opening in full gear position, as, in the Baker gear, the back port opening is larger than the front. This should be done with reverse lever at 50% cut-off position.

The foregoing applies to inside admission only, and any alterations in the eccentric rod length are just the opposite for outside admission valves.

CUT-OFF AND ECCENTRIC ROD.

The following rule will determine whether or not the eccentric rod is of the correct length: If the cut-offs are long on the front end in forward motion and long on back end in backward motion, the eccentric rod is too long. If the cut-offs come just the opposite of the foregoing, then the rod is too short. Valve setting may be considered correct when there is not more than ½ inch difference in the piston and cut-offs.

Valve Travel.

Place the reverse lever in full forward position and test the full valve travel shown on the drawings. Due care should be taken to see that the reach rods are the same length as shown on the drawings. If there is a difference between the right and left sides of the en-

gine, change the gear reach rod on the side of the engine where the short travel exists. After obtaining equal travel on each side of the engine in this manner, the reverse lever should be put in its central position and the main reach rod adjusted. Then the reverse lever quadrant length, or power reverse cylinder crosshead guide should be tested for the desired travel in both full forward and back motions. The full gear position of the reverse yoke for inside admission should not be less than 5⅞ inch from the bell crank pivot, in the forward motion, and not greater than 20¼ inches in the back motion. With outside admission gear, this dimension for the forward motion position of the reverse yoke should not exceed 20¼ inches, and not less than 6¼ inches in the back motion.

Eccentric and Valve Rod.

Place the engine on each of the four dead centers in full gear, both motions, and, with the valve stem tram, make the lead marks on the stem. The changes necessary on the eccentric rods and valve rods are determined by the lead marks, just obtained, by the following process. *On inside admission gears the direction of the change in the eccentric rod should be the same as that in which the valve should move to correct the forward motion.*

The eccentric rod adjustment is determined first, as follows: Subtract the smaller lead on one end from the larger on the same end and multiply the result by 2.

The valve rod error is determined as follows: Add the two leads on one end and the two leads on the other end, substract the smaller sum from the larger sum, and divide the result by 4.

The change in the valve rod is always in the direction of the greatest error. In the outside admission gear the eccentric rod change is made in the direction opposite to which the valve should move to correct the forward motion; otherwise the valve setting is the same.

Variable Lead.

When this setting is resorted to, adjust the valve gear the same as described for constant lead, except that the eccentric crank should be set accurately to the throw specified, and moved from that location only if the specified lead variation is not produced. It should be distinctly borne in mind that no adjustment of the eccentric crank can increase or decrease the total lead. With any change by that method, the lead in one direction is always obtained at the expense of the opposite motion. To give more lead to that direction of rotation in which the eccentric crank follows the crank pin the eccentric crank should be moved outward from the journal, or eccentric crank should be shortened, and in the reverse

direction, just the opposite. Having adjusted the eccentric crank location, eccentric rod length and valve rod length by the methods just explained, the valve setter should adjust the quadrant stops and reach rods to produce the specified maximum valve travel in each direction.

The cut-off should also be measured in at least one position of the reverse lever in each direction.

IN CASE OF ACCIDENTS.

The breakdowns most common to the Baker valve gear may be summarized as follows:

First—Those in which the main rod may be left in position without blocking the valves.

Second—Those in which the main rod is taken down and the valves must be blocked.

Third—Those in which the main rod may be left in position and the valves must be blocked.

With the main rod in position, it is necessary to have lubrication for the piston, and to relieve all compression in the cylinder, in another manner than by removing the cylinder cocks; that is, if there are relief, or vacuum, valves in the cylinder heads, these may be removed. This will prevent compression and also permit lubrication.

MAIN ROD IN POSITION AND VALVE NOT BLOCKED.

Suppose, for example, one of the followers are broken; an eccentric crank, eccentric rod, radius bar, reverse yoke, or upper end of the bell crank. Take down the eccentric rod and other broken parts, disconnect gear reach rod; block the lower end of the bell crank by using a U-bolt. This applies only to engines with the type shown in Fig. 104. The valve on disabled side then receives motion from the combination lever and the union link.

MAIN ROD REMOVED AND VALVE BLOCKED.

Considering the second of the above mentioned accidents: For example, the main rod is broken and piston rod bent; it is absolutely necessary to block the valve. Take down the main rod and remove the valve rod, secure the valve to cover the steam ports, by a small set screw in the side of valve rod crosshead guide, provided for this purpose.

If no set screw is provided a valve stem clamp may be employed. Clamp or block the main crosshead at the end of the stroke, and proceed with engine on one side.

MAIN ROD IN POSITION AND VALVE BLOCKED.

Regarding the third class of accidents: Let us assume that the valve rod or vertical arm of bell crank is broken.

Remove the valve rod and block the valve, as previously explained. If the combination lever or union link are broken, remove valve rod and tie the lower end of combination lever to clear the main crosshead, or remove the combination lever and block the valve.

With the above conditions care must be taken not to come to a stop with the main pin on disabled side on top or bottom quarter, as the combination lever would be perpendicular, and the valve would cover the steam ports. The crank pin on the opposite side would be on dead center, and it would thus be impossible to start the engine.

GEAR REACH ROD.

Should the gear reach rod break, block the reverse yoke and wire it securely so that it cannot move; otherwise, take down the eccentric rod.

MAIN REACH ROD.

Should the main reach rod break, block the reverse shaft arms and wire securely, or block and wire the reverse yokes.

In general, the best practice to follow, should any part of the valve motion fail, is to take down the valve gear parts that would interfere with the moving of the engine, cover the steam ports as described above, and proceed on one side.

THE SOUTHERN VALVE GEAR.

This form of valve gear was invented by Mr. William Sherman Brown, of Knoxville, Tenn., who made application for a patent on the gear July 11, 1911, which was granted to him July 23, 1912.

The inventor was employed as a locomotive engineer on the Southern Railway, and the first gear of this type was used on the Southern Railway Company's locomotive, No. 586, a 22x30-inch consolidated engine, employed in heavy freight service, which was turned out of the shop at Knoxville, Tenn., in February, 1913.

We understand the new gear proved a success from the start, and that the first engine on which it was applied, with the same gear and set of valves, was still in service after traveling 150,000 miles, with practically no expense for work done to the valve parts, which were replaced identically as they were removed when the engine was shopped for general repairs.

The fact that the Southern Railway Company has equipped a large number of their locomotives with this gear, after giving it a thorough trial in all branches of the service, is strong evidence of its merits.

Construction.

While this gear is a radical departure from all previous outside valve gears, it is a gear that can be adapted

to any class of locomotive, either inside or outside admission, and was designed with a view of eliminating round-house repairs, and delays to power incident thereto.

In designing the Southern valve gear, the inventor has eliminated many objectional features that have been found in other outside valve gears. This gear is very simple and compact, and contains but few wearing points. The crosshead connections have been dispensed with, and the gear has also been correspondingly reduced in weight.

In Fig. 105 we present the Southern valve gear, showing clearly the construction of the gear and the names of the parts. This, however, is an earlier type; the more modern form of the Southern gear is illustrated in Fig. 106, but the names of the parts are the same as those shown in Fig. 105.

If the valves are properly adjusted at the time the engine receives general repairs, the gear is so designed as to eliminate the necessity of any adjustment in the blacksmith shop, while the engine is in service. Simplicity of design has long been recognized as one of the greatest factors in the reduction of maintenance cost and the elimination of delays and repairs.

Transferring from a rotary to a reciprocating motion is accomplished by direct movements and on straight

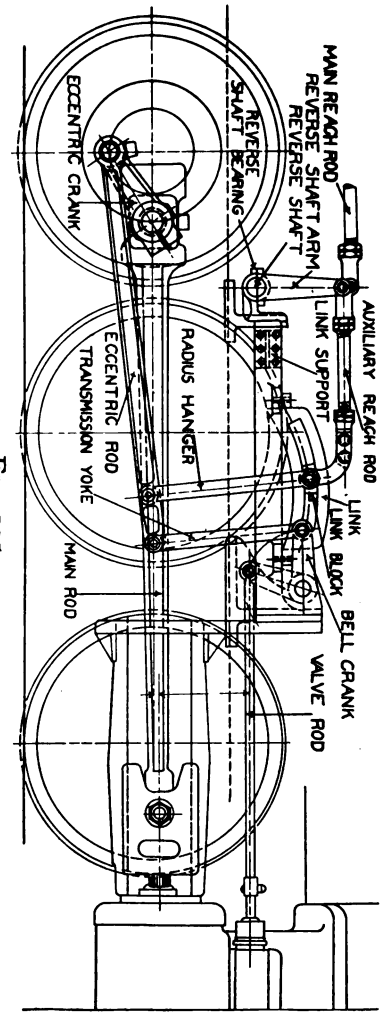

MAIN REACH ROD
REVERSE SHAFT ARM
REVERSE SHAFT
REVERSE SHAFT BEARING
ECCENTRIC CRANK
AUXILIARY REACH ROD
LINK SUPPORT
LINK
LINK BLOCK
BELL CRANK
VALVE ROD
RADIUS HANGER
ECCENTRIC ROD
TRANSMISSION YOKE
MAIN ROD

FIG. 105.

lines, thereby doing away with strains and distortions. The links are horizontal and stationary, which does away entirely with the wear at this point, as the block only moves in the link when the reverse lever is moved to adjust the cut-off, or reverse the gear. The links being stationary also does away with what is known as the slip in the link block, found in some outside gears. There are but eight possible points of wear to the side, or a total of sixteen wearing points per locomotive, this being less than half contained in some gears.

Fig. 106 shows a view of the modern gear applied to a 26x32 Santa Fe type locomotive. It may be seen that this gear differs in some respects from that illustrated in Fig. 105.

This gear will practically do away with engine failures due to breakage of valve gear parts. The different parts are so balanced as to reduce the wear on the pins and bushings to a minimum. A 22x30-inch consolidated engine, equipped with the Southern valve gear in heavy freight service, after making over 100,000 miles, did not show any appreciable wear on the pins and bushings; neither had there been one cent spent on repairs to the valve gear parts. All bearings, pins and bushings of the Southern valve gear are of such size as to insure long usage with very little wear. The forward end of the eccentric rod is supported by a bell crank

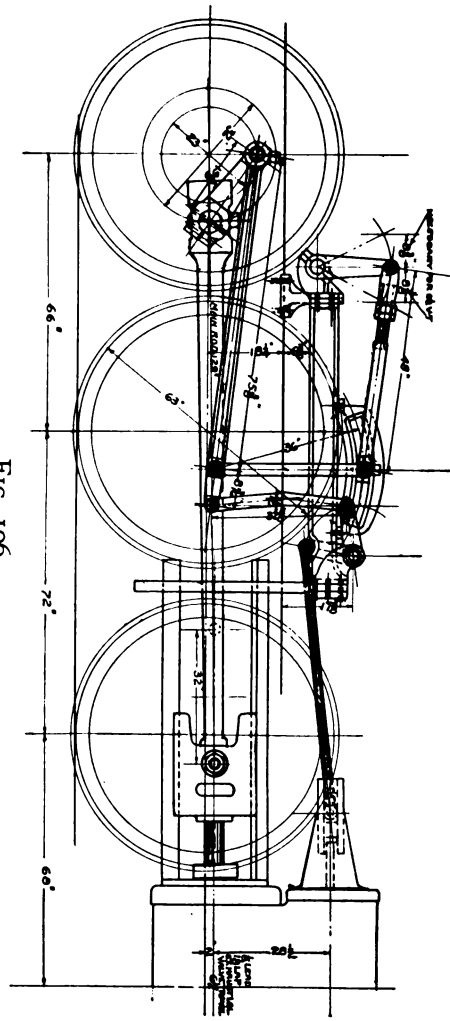


FIG. 106.

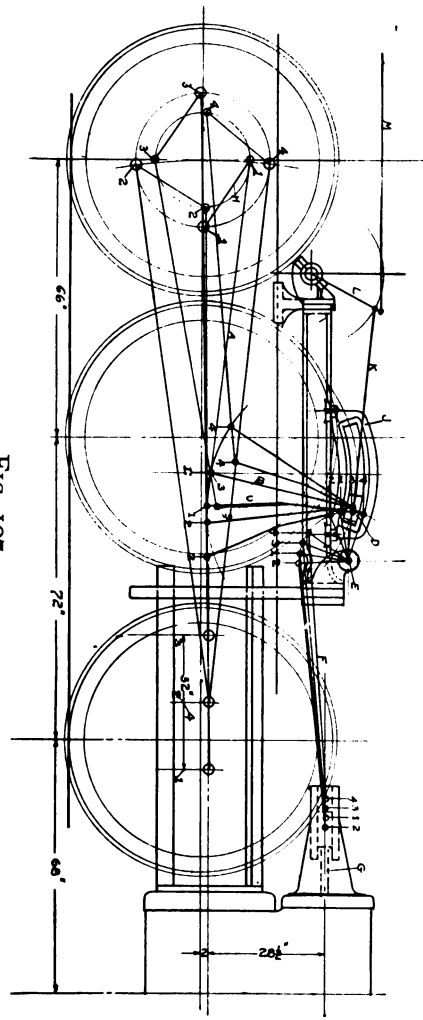


FIG. 107.

hanger, which has at its top two bearings spaced widely apart, thus absolutely preventing any side slap on the eccentric rod.

Fig. 107 is a kinematic diagram, showing the varying positions of the different parts of the gear in forward motion. Starting on forward dead center, or on position Nos. 1—1 and C, moving to position Nos. 2—2 and C, bell crank then returning back over position No. 1—1 to position Nos. 3—3, thence to positions No. 4—4, and then to position Nos. 1—1, we follow the diagram of the gear through the entire stroke.

The Southern gear is designed to eliminate all stress and strains on reverse lever and reach rod connections, and the reverse lever is easily handled while working full head of steam. This feature appeals strongly to the engineers, enabling them to adjust their cut-off without fear of the lever getting away from them, and will induce them to work at as short cut-off as possible, resulting in a saving in fuel.

Fig. 108 is a kinematic diagram showing varying positions in back motion. Starting on forward dead center, or position Nos. 1—1 and C, moving to position Nos. 2—2 and C, then bell crank returning back over position Nos. 1—1 and C to position Nos. 3—3 and C, thence to position Nos. 4—4 and C, from which it moves to position Nos. 1—1 and C, the stroke is completed.

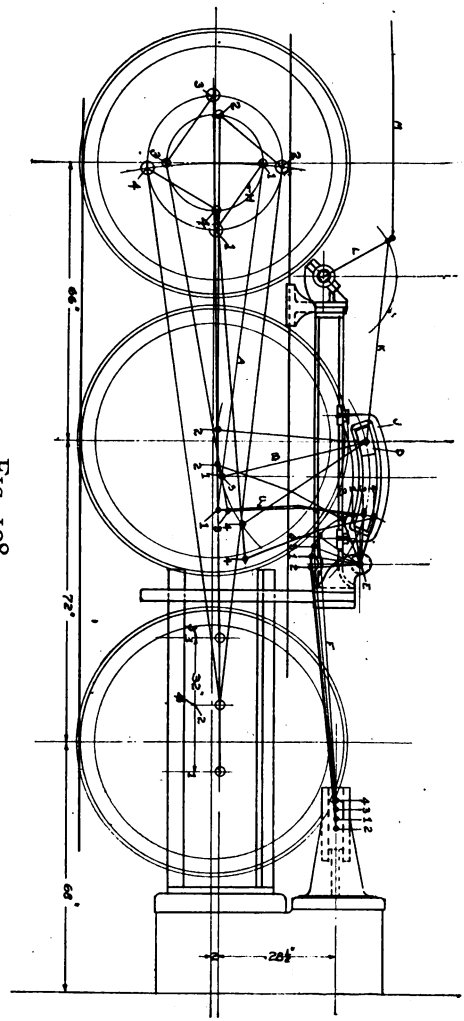


FIG. 108.

Directions for Setting and Adjusting Southern Valve Gear.

The method of setting and adjusting the Southern gear is clearly explained by the following rules, which have reference to the Southern valve gear layout illustrated in Fig. 109.

First. Set the link support so that dimensions conform to figures on erecting card. Then connect the gear as shown on the erecting card.

Second. Set the *links* so that dimension (M) conforms to erecting card.

Third. Set reverse lever in center of quadrant, adjust the main *reach* rod, so that *reverse shaft arms* will stand in vertical position.

Fourth. Adjust *auxiliary* reach rods so that *link block* will be in *center* of link when the reverse lever is in center of quadrant.

Fifth. Set *eccentric crank* for outside admission to lead *main crank pin,* and for inside admission to *follow main crank pin,* at a distance from center of main axle to center of *eccentric* crank pin to conform to erecting card.

Sixth. Find dead centers in usual way. With the engine on *front* dead center (F), tram from center of radius hanger pin (P) to any stationary point on cylin-

der casing or guide yoke, and scribe arc (a) as shown. Then place engine on *back* dead center (B) and again scribe arc (b) from center (P). If arcs (a and b) fall line and line, the eccentric crank setting is correct. If they do not, then knock eccentric crank to or from center until they do.

Seventh. Then revolve wheel or move engine one full turn, and tram in same center (P) and scribe arcs (c and d). In full travel (N) is within 1/16 inch of eccentric rank circle (O) shown on erecting card they are correct. If there is a difference of as much as ⅛ inch full travel of eccentric crank pin, lengthen the eccentric crank having the least *full* travel *one-fourth* of the *difference,* and reset cranks as per paragraphs 5 and 6.

Eighth. Place engine on *front* and *back* dead centers (F) and (B) and move reverse lever entire sweep of quadrant. If valves move in same direction as link block, move link ahead, if in opposite direction move the link back until still valve is found.

Ninth. To get the length of the valve rod, place engine on front *dead* center, allow ⅛ inch for lead, or amount shown on erecting card and adjust valve rod to lead. (In setting engines on jacks, have the wheels placed ½ inch under actual running height; for instance, if the blue print running height is 18 inches

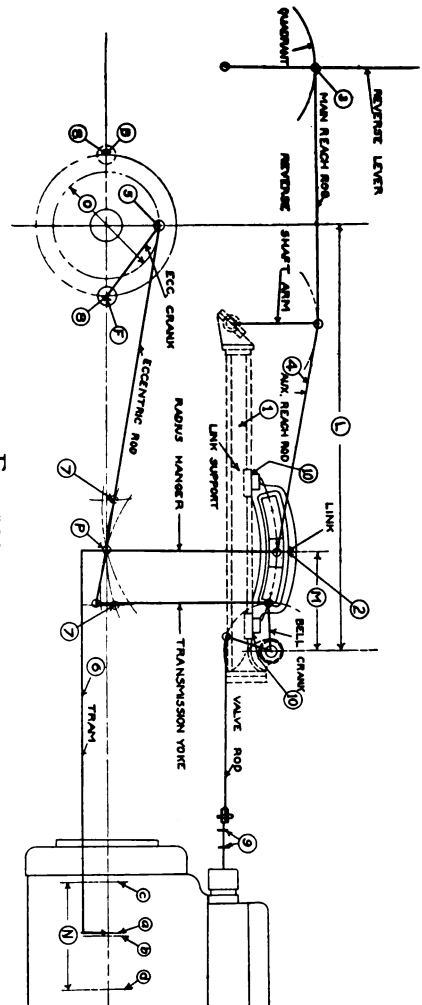

QUADRANT
REVERSE LEVER
MAIN REACH ROD
REVERSE SHAFT ARM
ECC. CRANK
ECCENTRIC ROD
AUX. REACH ROD
LINK SUPPORT
RADIUS HANGER
LINK
BELL CRANK
TRANSMISSION YOKE
VALVE ROD
TRAM

FIG. 109.

when the engine is on the rail, set the main wheels at 17½ inches running height on jacks.) Then try for full valve travel in forward position on each side of engine. If the valve travel is found to be unequal make necessary corrections by lengthening or shortening the auxiliary reach rods. For example, say 6 inches is the desired valve travel, lengthen the auxiliary reach rod on left side until 6-inch travel is obtained. Then proceed to run cut off in full gear, and if necessary lengthen or shorten valve rod to equalize cut off. Then run cut off at about 9-inch piston travel and equalize same.

Tenth. Raising or lowering the links, by means of liners, has the same effect as shortening or lengthening valve rods, therefore any derangement which may occur can be corrected without disconnecting any part of the gear.

Breakdowns.

First. If the eccentric crank or eccentric rods fail, disconnect the eccentric rod from the crank, radius hanger and transmission yoke, tie up the hanger and the yoke, clamp the valve in its central position, and proceed.

Second. If the radius hanger fails, disconnect the hanger from the rod and take down the eccentric rod. Then clamp the valve in its central position, and the locomotive is ready to proceed.

Third. In the event of the transmission yoke failing, disconnect it from the eccentric rod. Then proceed, first clamping the valve in its central position.

Fourth. When the horizontal arm of the bell crank fails, disconnect the yoke from the eccentric rod, tie up to clear, clamp the valve centrally, and proceed.

Fifth. If the vertical arm to the bell crank fails, or breaks, clamp the valve in its central position and proceed. It will sometimes be found necessary, however, to take down the broken vertical arm.

Sixth. In case of the failure of one auxiliary reach rod, or the reverse shaft arm, block both link blocks in the same position in the links, and in such a position as to give sufficient port opening to start the train and to control its speed by the throttle.

Seventh. If the main reach rod, or the middle arm to the reverse shaft, fails, or if both auxiliary reach rods fail, block the link blocks same as in case of one broken auxiliary reach rod, or reverse shaft arm. The power and speed of the locomotive may then be controlled by the throttle.

Laying Out the Southern Valve Gear.

By J. B. Gwin, Chief Draftsman of the Southern Locomotive Valve Gear Co., Knoxville, Tenn.

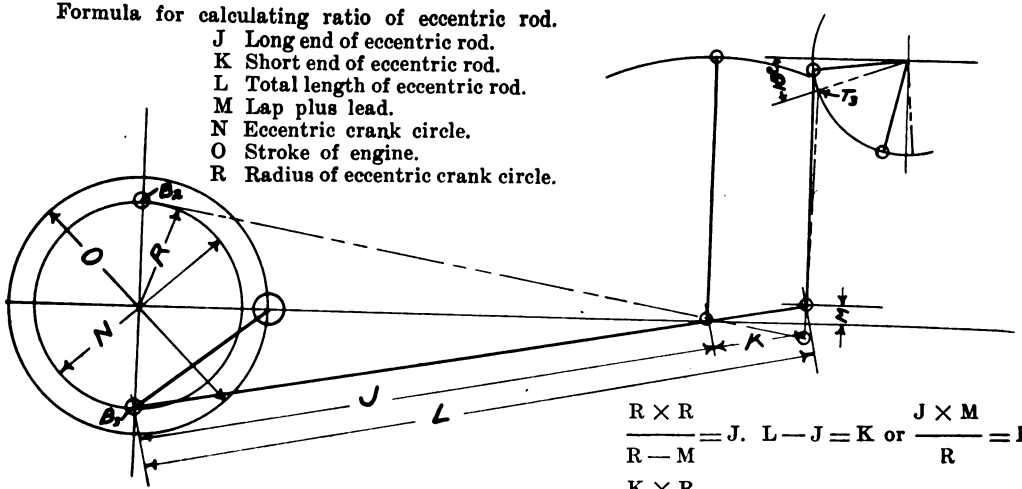
Lay off the diagram on scale of half size, as follows: Lay off the center line through the center of the drivers,

Fig. 111, and cut this line with a vertical through the center of the main axle at A. From center A, lay off both main pin circle, and eccentric pin circle, d—d and e—e respectively. Now locate the center of cylinder; then, with tram set to the length of the main rod, locate the center of the crosshead pin on the center line of the cylinder at 1, from forward dead center. Now, with a straight edge through centers A and 1, draw the center line of the main rod. This is done in order to get the exact position of the main pin center above the center line of the drivers, which is very necessary for obtaining the length of the eccentric crank. Next, locate the bell crank center T to suit the available horizontal distance from the center of the main driver, and the vertical height above the center line of the drivers, if possible, so that the pin in the vertical bell crank arm will stand ½ inch below the center line of the valve stem, with the pin in its lowest position. Then, with radius equal to the length of the bell crank arms, from bell crank center T scribe arc T_1. Next drop a vertical from a point T_3, in Fig. 110. Then the length L measured on diagram, Fig. 110, will be the approximate length of the eccentric rod.

Now calculate the ratio of the long and short end of the eccentric rod, using diagram formula, Fig. 110. Set the tram to the equal length of the long end of the eccentric

Formula for calculating ratio of eccentric rod.

J Long end of eccentric rod.
K Short end of eccentric rod.
L Total length of eccentric rod.
M Lap plus lead.
N Eccentric crank circle.
O Stroke of engine.
R Radius of eccentric crank circle.



$$\frac{R \times R}{R - M} = J. \quad L - J = K \text{ or } \frac{J \times M}{R} = K$$

$$\frac{K \times R}{J} = M.$$

FIG. 110.

rod, and lay off arcs b—b, from centers B and B_1, intersecting at E, the center of the lower radius hanger pin when the engine is standing on either dead center.

Now, with the same tram, lay off arcs b_2—b_3, from centers C and D; then set the tram to the total length L of the eccentric rod, and lay off the arcs f and f_1, and the arcs C_1 and C_2, from the centers B and B_1, and C and D, to the extreme forward centers of the lower transmission yoke pin centers F, F_1, F_2, F_3, F_4, and F_5. Now set the tram to the length equaling the height of bell crank center T above center line of drivers, plus 1⅜ inches; this will be the suitable length of the radius hanger. For example, say that the height is 33⅛ inches, plus 1⅜ inches, which would equal 34½ inches, the length of the radius hanger. Now strike the arc G, from the center E; arc G will be the radial center, and the intersection of the arc G and the heavy line J_1 will be the vertical center, of link and radius hanger. Now lay off the centers H and I. This distance will have to be found by trial, in order to obtain the proper valve travel. I is the link block center position, forward motion, and H is the same, for back motion. Take the tram with length equal to the radius hanger, and, from center I strike the arc J, also from center H, strike arc K. Now lay off the eccentric rod center lines as follows: Lay off the center a, cutting centers B and E, and arc f at F.

This for inside admission valves, and a_1 for outside admission valves, cutting centers B_1 and E, and arc f_1 at F_1. Lay off a_2, cutting center A and arcs b_2 and C_1, and J at L and F_2. Lay off a_3, cutting center A and arcs b_2 and o_1, and K at O and F_3. Lay off a_4, cutting center A and arcs b_3 and C_2, and K at N and F_4. Lay off center a_5, cutting center A and arcs b_3 and C_2, and J at M and F_5.

Now set tram to a trial length for the transmission yoke, say about 3¼ inches shorter than the radius hanger. From center F_2, scribe the arc l, cutting the arc T_1. And from the center F_4, scribe the arc m cutting the arc T_1.

Now from the centers F_3 and F_5, scribe the arcs n and o, cutting the arc T_1.

Now, if the arcs l, m, n, and o intersect exactly on the arc T_1, this shows that the eccentric rod and transmission yoke are of proper length. But if the arcs l and m intersect on the arc T_1, and the arcs n and o intersect on the outside of arc T_1, the transmission yoke needs to be lengthened.

If the arcs n and o intersect on the arc T_1, and l and m on the outside, the transmission yoke needs to be shortened. But, if all four arcs l, m, n, and o intersect an equal amount outside of the arc T_1, the eccentric rod needs to be lengthened, while, if all four arcs intersect inside of the arc T_1, the eccentric rod must be shortened.

It is needless to say that any change in the length of the eccentric rod will call for a revised calculation for the lap and lead ratio, by the diagram formula, Fig. 110.

For inside admission, the main pin leads the eccentric crank pin, and with outside admission, the main pin follows the eccentric crank pin. The full black lines of the eccentric crank and rod are for inside admission, and the broken lines of the eccentric crank and rod are for outside admission.

THE YOUNG VALVE GEAR.

A new design of radial outside valve gear, designed to take care of the large cylinder volumes now being used, which is really a modification of the Walschaert valve gear, and somewhat resembles the Lewis valve gear which was used on a few roads between 1892 and 1895, was patented September 2, 1913, by Mr. Otis W. Young, of Chicago, Ill.

The valve gear was designed to give better and more economical steam distribution, with increased valve travel, than is possible to obtain with the use of the Walschaert valve gear. But with the same travel, lap, lead and clearance as the Walschaert valve gear, the Young gear gives no better events than the Walschaert type.

Within the limit of 6½ inches valve travel, the Walschaert valve gear performs satisfactory service, but with 7 inch travel or more, common to the larger and more modern locomotives, very objectionable angularities arise. These are eliminated by the use of the Young valve gear, which gives satisfactory results with a valve travel up to 8½ inches. The object of greater valve travel than has heretofore been practicable is to increase the lap and lead and consequently provide wider port openings needed for handling large cylinder volumes.

The equipment is very simple in construction, embodying rigid bearings, and straight thrusts without objectional angularities. The entire arrangement is well standardized, so that new parts are easily obtainable.

Design and Construction.

The Young design is an entirely new form of radial type of outside valve gear. Its simplicity of design and construction is due to the embodiment of the fewest possible working parts.

The motion of this valve gear, as may be seen from the illustrations, Figs. 112 and 113, is derived entirely from the piston and crosshead connections, thereby eliminating distortions resulting from the slip of the driving boxes, wear of journals brasses, and settling of the equipment upon its springs. The 90 degree movement of the main travel is obtained by a pair of con-

centrically mounted rockers. These rockers transmit the variable component of the motion from the crosshead and link on one side to the lap and lead lever on the other side. A reduction in the size of the block and the link is made possible by constructing the link with integral trunnions, thus obtaining a reduction of 40 per cent in reciprocating weight, without sacrifice to wearing surfaces. Fig. 112 shows the Young valve gear, with the steam chest cut away to show the Young piston valve in operation.

Each piston, in operation, causes valve movement equal to the lap and lead on its own side and travel on the opposite side. The valve is moved a distance equal to its lap and lead each way from its central position by the movement of the piston through an entire stroke, when the radius bars are both central in the link. In this valve gear the source of motion is the piston alone, and the right engine is 90 degrees ahead of the left; while the left engine is 270 degrees ahead of the right, therefore the valve travel producing members must be direct on one side and indirect on the other. Thus, for forward motion the back end of the right radius bar is above the center of oscillation of the link and the left radius bar below. The reverse is true for backward motion. The various cut-offs are obtained with the radius bar in intervening positions between its central and ex-

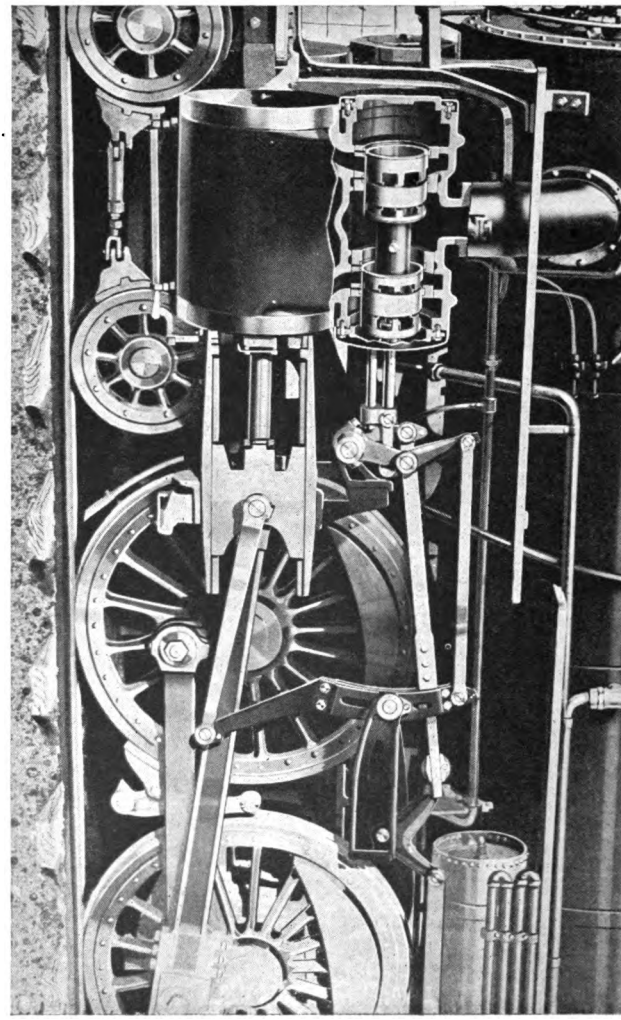


FIG. 112.

treme locations. Valve travel in addition to lap and lead is produced through the oscillation of the link which causes the radius bar to oscillate the rock shafts. This motion is effective on the valves on the side of the locomotive opposite to the link imparting motion. As the front end of the radius bar acts on the lap and lead lever between its two extreme connections, unusually long travel is imparted to the valve without excessive link angularities. As the movement of the links is constant under all operating conditions this is desirable.

Valve Setting.

In locomotives using valve gears that derive a portion of their movement from a connection to the driving wheels, as, for instance, the Walschaert, it is necessary to take wheel centers before proceeding to set the valves—this is essential for the reason that only by knowing the correct wheel centers, is it possible to determine whether adjustment is needed in the eccentric crank or in the eccentric rod.

With the Young valve gear, which has no connection to the driving wheels, the method of taking wheel centers may be simplified. It can, and, in fact, is, pre-determined on the drawing board, by the following process. The length of the main rod, and the piston stroke, are known in every valve application, and it is only a matter of laying down a few lines on paper to determine

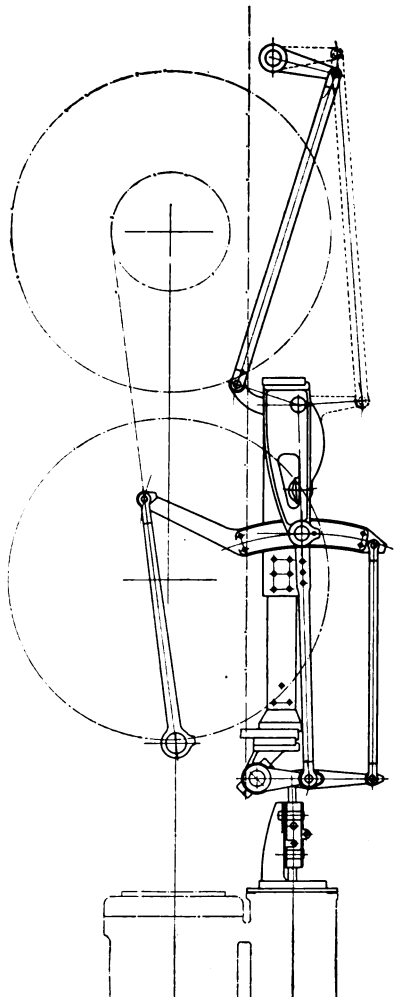


FIG. 113.

how far back of its central position one crosshead must stand, due to the angularity of the main rod, when the other crosshead is at its extreme travel, which position it must occupy when the wheel is at its dead center. All erecting cards showing the assembled Young valve gear are arranged, therefore, to show the exact distance back of its central position that one crosshead will occupy, when the wheels on the opposite side are on their true centers. This distance varies in different designs of locomotives if the main rod lengths and piston stroke vary, consequently it must be obtained from the blue-print for the class of locomotives on which the Young gear is to be set.

For these reasons it is not necessary, in setting the Young gear, to take wheel centers in the shop, or even to have the wheels under the engine, nor is it necessary that the main rods be connected; in fact, it is better not to have the main rods up.

With the above thoroughly understood, the valves may be set as follows:

With crossheads connected to pistons, the main rods disconnected, mark the piston striking points on the guides.

Show a mark on the guides central between the striking points, place the crossheads back of this central mark the distance designated on the erecting card.

If the radius bar can then be moved from its lowest to its highest positions, without movement of the valves, the dimension on the union link is correct. If not, shorten or lengthen the union link to suit.

With one crosshead in the above position, move the opposite crosshead to each end of its stroke, and equalize the lead to show the same opening for front and back ports, by means of the nuts on the valve stem.

Repeat the process on the opposite side. Then adjust the main rod lengths to obtain equal piston clearance in both ends of the cylinder.

The valves will then be perfectly square, without resorting to rollers for checking the cut-off.

Breakdowns.

Because of its simplicity of construction and operation, and its rigid bearings, the Young valve gear is much less liable to breakdowns than is the Walschaert gear. An objection is often made, that in the event of a break down on the road, a locomotive equipped with the Young gear is unable to proceed after one side is disconnected. This is true to a certain extent, but it must be remembered that cases are few where modern large locomotives have actually been brought in under their own power after disconnecting one main rod; therefore the Young valve gear is as efficient in this respect

as the other forms of valve gear in use today. In fact, one valve may be disconnected, and the ports covered, and as long as both main rods are up, the Young gear will continue to operate. To meet this objection, however, an arrangement may be provided wherein the union link connects the reversing link to the main pin instead of to the wrist pin. This arrangement has an advantage over other valve gears inasmuch as it eliminates the use of an eccentric crank. It provides the same valve motion as the connection to the wrist pin, but is not, however, the preferred method.

This valve gear is manufactured by the Pyle-National Company, of Chicago, Ill.

THE END.

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